for . . .

engineers and contractors controlling the design, specification, installation and maintenance of heating, ventilating and air conditioning.

HEATING & VENTILATING

AIR CONDITIONING

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EDITORIAL STAFF

Editor
CLIFFORD STROCK

Associate Editor
JOSEPH F. KERN, JR.

Contributing Editor
C. H. B. HOTCHKISS

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REPRESENTATIVES

Eastern Representative
JOHN F. HYDE
148 Lafayette Street
New York, N. Y.

Western Representative
GEORGE G. TURNER
228 North La Salle Street
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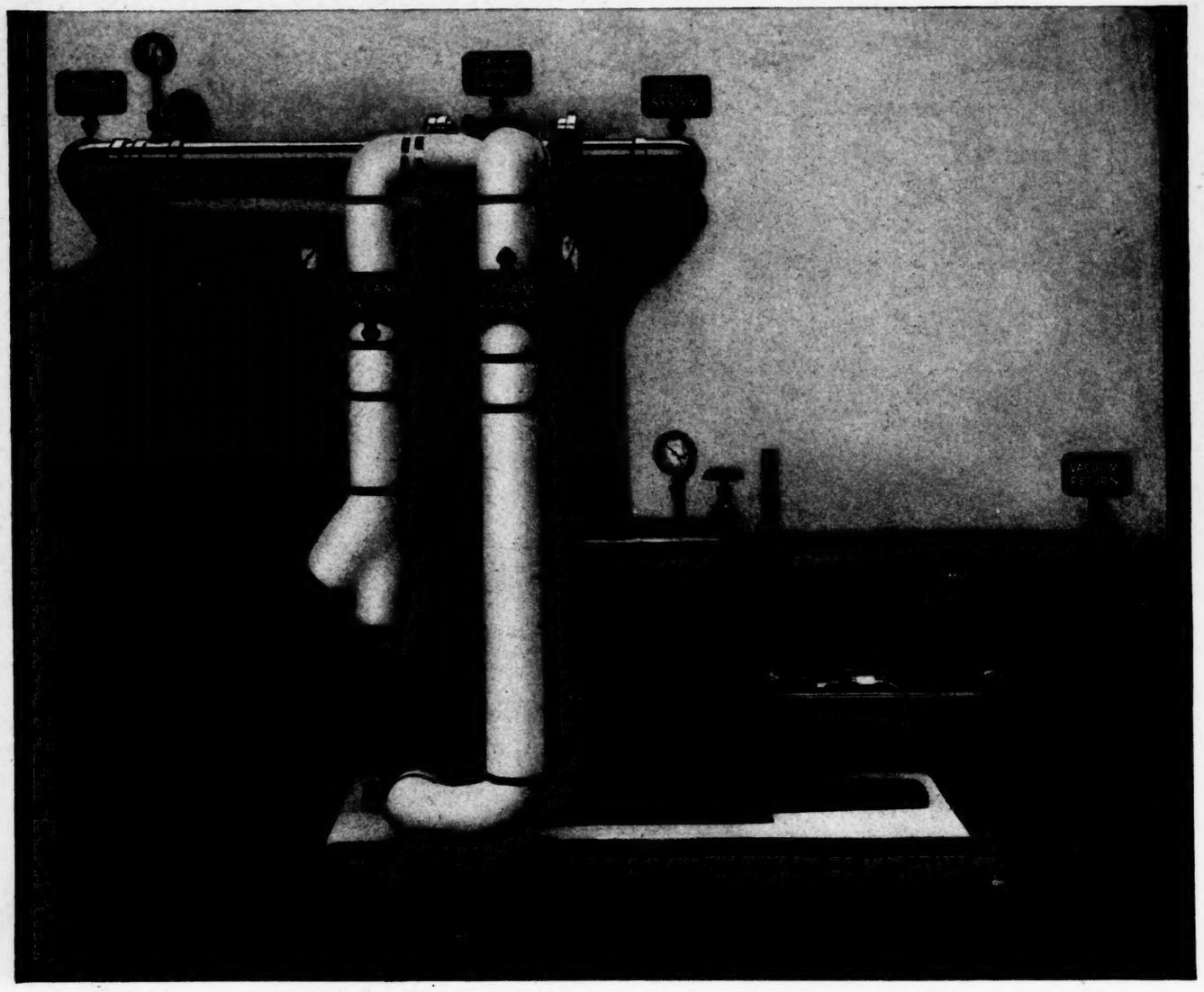
Where Heat Must Not Fail— Install this SAFE Heating Pump!



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

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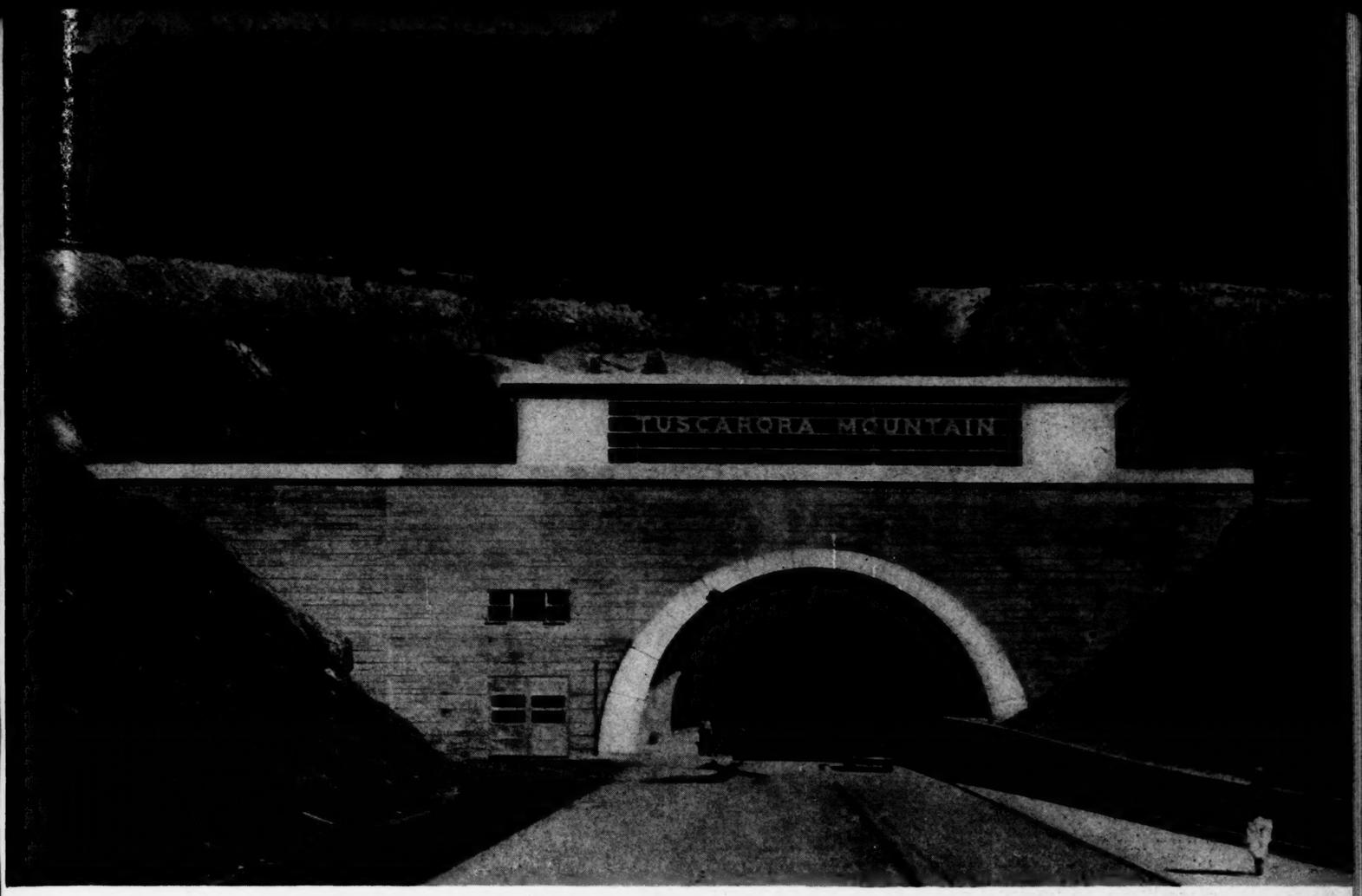


Fig. 1. The west portal of the Tuscarora Mountain tunnel of the Pennsylvania Turnpike. For a view of the road itself, see front cover.

Ventilating Pennsylvania's Super-Highway

Opened on October first, the 160-mile Pennsylvania Turnpike linking Harrisburg and Pittsburgh has seven ventilated tunnels, originally built by Vanderbilt in the Eighties for his proposed New York-Pittsburgh railroad. The tunnels have a total length of nearly 7 miles and are provided with automatic control of CO. The accompanying descriptions are devoted to, first, the ventilation of the tunnels and, second, the heating and ventilating of the service buildings. The second part was prepared by Roderick A. Wood.

SIX million *more* cubic yards of excavation were required in construction of the Pennsylvania Turnpike than were dug in the construction of the famed and ill-fated Maginot line. Almost the same amount of steel for reinforcinig was also required, 46,500 tons against the Maginot's 50,000. The Turnpike was built by more than 155 contractors from eighteen different states. Its 160 miles of concrete lanes tunnel under the Alleghenies at seven different points the great mountain barrier between the Middle West and the Atlantic Seaboard.

Planned for safe driving at high speeds, the Turnpike has two wide, reinforced concrete lanes in each direction, separated by a grass plot center. There are no grade crossings, but side traffic into and out of it is provided for by eleven interchanges equipped with accelerating and decelerating lanes. The maximum curvature is six degrees.

The maximum grade of 3% is made possible by seven ventilated tunnels totaling nearly seven miles in length, which save nearly 10,000 feet of lift in crossing the mountains. On each side is a 10 ft. shoulder onto which cars can pull off for stops. Total width of the roadway is 78 ft. and minimum width of the right-of-way 200 ft.

The ventilated tunnels are lined with concrete and reinforcing steel. In some cases where the roof of the fifty-year old bore had fallen, heavy loads of rock were filled in above the steel and concrete arches to take care of the voids. In constructing the tunnels a false roof of 5 in. concrete slab was built across each arch and supported by stainless steel hangers, the space between the slab roof and the arch forming a duct for the conveyance of fresh air in ventilating the tunnels.

Each tunnel has a two-lane roadway 23 ft. wide between curbs, with a minimum of 14 ft. 4 in. overhead clearance; a sidewalk on one side, 2 ft. 10 in. wide; a ledge on each side which forms a casement for low tension ducts carrying control, communication and signal cables; a bank of high tension cables under the sidewalk carrying power cables for the ventilating equip-

Fig. 2 (left). The condition of one of the tunnels as it was prior to the Pennsylvania Turnpike Commission beginning work. In Fig. 3 (right) is shown a similar view for an almost completed tunnel.

ment; a water line for fire and flushing; and various drainage lines.

During excavation and construction work, ventilation for workmen was provided by blowers of sufficient capacity to maintain a concentration of below 1,000,000 particles of dust (under 10 microns in size) per cubic foot of air when drilling in rock containing less than 10% by weight of free silicon dioxide; and under a concentration of 10,000,000 silicon dust particles per cubic foot of air when drilling rock having a free silicon dioxide content of 10% or more by weight. Some ventilation units could either blow or exhaust, the latter being used immediately after blasting. Ventilation pipe, used when cutting the tunnels in all except one, were of spiral riveted steel. In the exception, the contractor constructed the drainage gallery as he moved along and used it both for drainage and ventilation.

Fig. 4. One of the large Sturtevant tunnel fans. It is reported that these fans are the largest on which efficiency tests were run.

Permanent Tunnel Ventilation

Permanent ventilation of the tunnels is accomplished by Sturtevant blower type fans, among the largest ever built for this type of service; in fact the largest ever tested at the Sturtevant plant, all of the same general design and diameter. Variations in total air pressure and volume required for the different lengths of tunnel are accomplished by variations in speed to suit the different conditions.

Each tunnel except the shortest has four fans, this exception having but two, making a total of 26, each of 110 in. wheel diameter. These are housed in specially-constructed buildings of native stone over the top of each tunnel portal, two at the end of each tunnel.

The grand total air volume, contract rating, is 366,600,000 cu. ft. per hr. Largest volume for any one fan is 313,000 c.f.m., varying from that down to 185,000 c.f.m. for the smaller tunnels, the volume depending upon the length of bore. In other words, fan speeds vary so as to produce a volume of approximately 190,000 c.f.m. at each one inch of water gauge for the Blue Mountain tunnel to 311,000 c.f.m. with approximately 4 inches water gauge for Ray's Hill tunnel. In the case of the Sideling Hill tunnel the 313,000 c.f.m. is with about 2 inches water gauge. The tunnel with only two fans has a housing at one end only. The individual fan wheels have a double inlet width of 6 ft. 4 in., the fan shaft being 10½ in. in diameter at the wheel hub and tapered to bearing diameters at the ends, with total 19 ft. 6 in. shaft length.

Since all the fans are the same diameter, it was only necessary to test one of them to fulfill contract obligations. One of them was set up at the manufacturer's plant and tested according to standard test codes. Test requirements included setting up an outlet tube 12 ft. 9 in. in diameter and about 130 ft. long, with a 24-hour run.

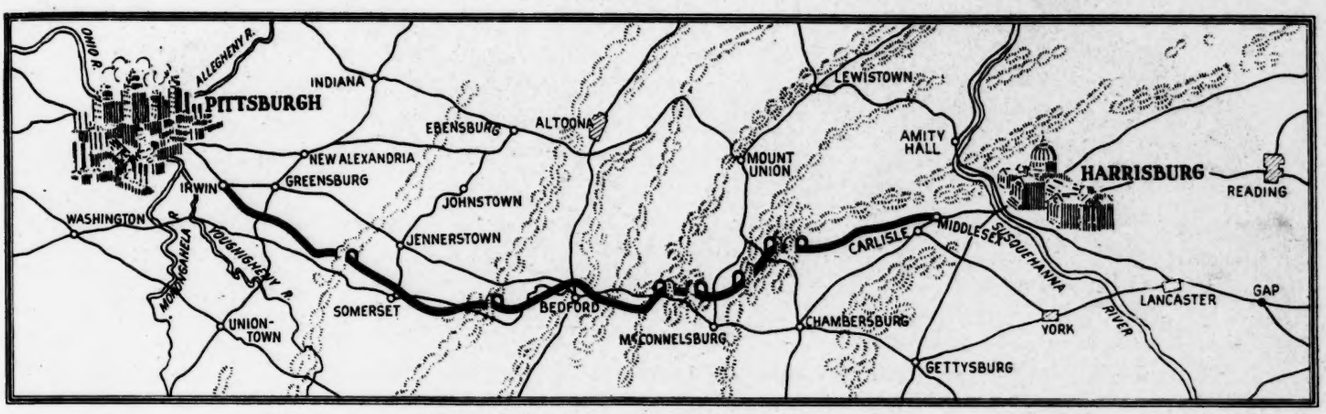


Fig. 5. Route of the Pennsylvania Turnpike showing location of the tunnels.

At each tunnel portal, in the fan building, is an analyzer and calibrator, which draws samples of air from various points in the tunnel, each analyzer serving one-half of each tunnel, except for the Ray's Hill tunnel, where samples are drawn to the west portal from the entire length of the tunnel.

Filters are installed in niches cut into the walls at the points where the air samples are drawn, to eliminate dust and other particles from the air as it is brought to the analyzers by vacuum pumps. Each analyzer reports its analysis automatically by wire to a recording instrument near the supervisor's station—a supervisor for each tunnel, except for the two Blue Kittatinny mountain tunnels, where one supervisor handles ventilation controls for both tunnels. This supervisor has four analyzers and recorders to observe and check.

Ventilation is automatically controlled to keep the carbon monoxide content in all tunnels at less than four parts in 10,000 of air at all times. As CO content varies, the supervisor is able to control operation of fans in a given tunnel for his station. Thus, if a fan operating at quarter-speed can keep the CO concentration down to the desired point, he can press a button to lower the fan operation to quarter-speed in any fan he may select. There are also buttons for half-speed and return to full speed.

Fan capacity was determined on an assumed rating of 175 cu. ft. of air per ft. of tunnel length, at full speed of all fans in operation. It is estimated, however, that full speed will be required only a very small fraction of one per cent of the time, giving a good margin of safety. Each fan has two motors, those for half and quarter speed of the two-speed type and for full fan speed of the single speed type. They vary from 40 hp. at the Blue Mountain tunnel to 250 hp. per fan motor for Ray's Hill.

Damper control is automatic, tied in with the starting cycle of the fan motors, each damper having a motor speed reducer and shaft with drums for winding cable that connected to the dampers, operation of which is in a pre-determined cycle according to the accelerating and decelerating conditions of the fans and drives.

Sources of Power

Power for the tunnel ventilation, portal buildings, ticket offices and maintenance buildings is provided by the different power companies in the various areas. Each tunnel has a sub-station equipped with single phase transformers having primaries suited to the various incoming voltages, with secondaries on each transformer to provide 2300 volts and 400 volts as required. Provision has been made for a fourth transformer spare if ever needed. Power distribution from the tunnel substations is controlled by the supervisor. Fan motors up to 100 hp., operate on 440 volts and those of 100 hp. and above on 2300 volts.

Utility Buildings

Each utility building at the tunnel portals has a separate heating plant, together with private facilities for

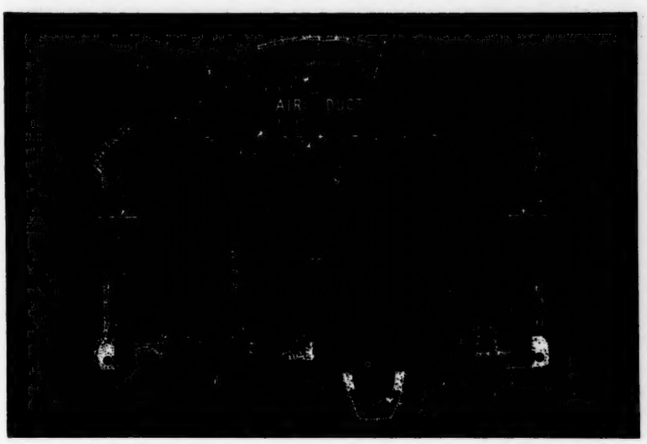


Fig. 6 (left). A section of a typical tunnel showing the space in the upper part used for the air duct. Fig. 7 (right) shows a photograph of one of the tunnels in an uncompleted form. This also shows the location and relative size of the air duct.

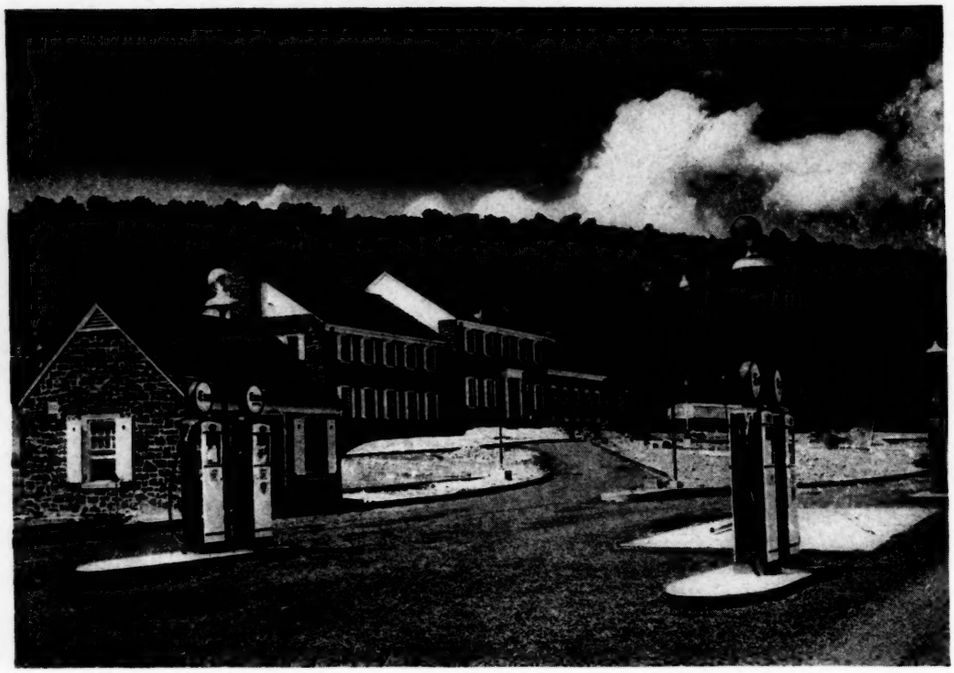


Fig. 8. In the center background is shown the Midway Main building while in the left foreground is the service station. The Midway Main building provides accommodations for tourists and truckmen. Service stations are located at eight other intervals along the right-of-way.

attendants, 5 kw. generator for emergency use, printing counter assembly and space for radio equipment for some future time. Low pressure heating boilers, one for each building, are used in the utility and tunnel portal ventilation buildings. Firing in the latter will be done by automatic stokers, using anthracite for the eastern tunnels and bituminous for the western ones. Ash will be disposed in pits with capacity for about two months' accumulation.

Each ticket booth is heated by a unit in the ceiling connected through an overhead canopy with the small heating plant in the utility building. Side walls of the booth are of heat absorbing glass, set at angles to eliminate reflections that might bother either ticket man or motorist. Outside walls are porcelain enamel on 16-gauge steel sheets which are removable and backed by one inch of insulation.

Tunnel ventilation was designed by Harold F. Hagen, B. F. Sturtevant Company. Sturtevant sub-let the equipment installation to the following Pennsylvania contractors: Capitol Construction Co., Hollidaysburg; Navarro Corp., Pittsburgh; Boyd H. Kline, Bloomsburg; R. S. Noonan, York; Ritter Bros., Harrisburg; J. F. Drass, Hollidaysburg; Snively Bros., Pittsburgh; and Wm. M. Clark Co., Newcastle.

Service Buildings

Two-pipe hot water heating serves the ten buildings owned by the Standard Oil Company of Pennsylvania along the Turnpike. These ten buildings consist of one large midway main building, near the half-way mark on the Turnpike, where motorists may eat and where truckers may stop to sleep, and nine automotive service stations at regular intervals on the Turnpike. All ten buildings are heated by oil.

Most interesting of these buildings from the heating point of view is the large midway main building, which is divided into three heating zones, each regulated by a thermostat connected to individual hot water circulators. These three zones are: (1) Basement and most of first floor, including lobbies, kitchen, and dining room, as well as second floor dormitories; (2) Part of first floor, including manager's office, barber shop, lunch rooms, and truckmen's lounge, and (3) the manager's living quarters on the second floor. A fourth circulator pumps water to a service station adjacent to the main building, and a fifth circulator serves another service station on the opposite side of the highway through pipes going under it.

An accompanying diagram shows heating details of the basement and first floor of the Midway Main Building. The second floor is not shown.

A 5000-gallon fuel oil tank supplies No. 3 oil to the Midway Main Building's Arco boiler, rated at 3700 sq. ft. EDR and carrying 2832 sq. ft. of radiation, of which 2275 sq. ft. are in the Midway Main Building itself, 188 sq. ft. in the adjoining gas station, and 369 sq. ft. in the gas station on the opposite side of the highway. The boiler in the Midway Main Building is fueled by a Gilbarco oil burner.

The seven remaining gas and service stations on the highway, spaced on alternate sides of it at 20-mile intervals, each has a heating load of 547 sq. ft. EDR carried by a boiler rated at 1067 sq. ft. EDR. These smaller boilers are Gilbarco boiler-burner units, burning No. 3 oil.

Minneapolis-Honeywell controls on these boiler-burner units include an immersion aquastat, protectorelay, protectostat, and a 250-gallon instantaneous submerged coil water heater.

Design conditions for the Midway Main and other buildings were for heating to an average temperature of 70F, with a maximum wind velocity of 15 miles per hour and an outside temperature of 0F.

In keeping with the progressive methods and high-grade equipment used in heating these new buildings is the cabinet type of convector, recessed and totally concealed, used in all ten of the structures of this de-

velopment. Just above the radiator surfaces is an air chamber, connected to a ⅛ in. pipe leading to a ⅛ in. air vent, almost at window sill height, which is operated with a screw driver through the top grille. This arrangement is convenient since it obviates stooping and it eliminates the use of a special and often-elusive key for operating the air vent.

Two designs of recessed heaters are used: One for outside walls, and one for inside partitions. The heater for the outside wall has its exit grille at the top of the recess, in the window sill, so as to provide a wiping of hot air on the cold window surfaces above it; the partition heater has a side air exit grille, with the entrance grille set at floor level. Tuttle & Bailey grilles were specified for both types of recessed heaters.

Among other sound constructional features embodied in these buildings is the location of control valves, which are installed on main branch lines from the boiler header before the flow valves, and on the returns behind the circulating pumps.

Ventilation of the midway main building is accomplished, in the basement and part of the first floor, by the exhaust duct system shown in an accompanying diagram. This system is connected to an exhaust fan with backward curved blade, direct connected and mounted on the ceiling, drawing 1600 c.f.m. at 1 in. static pressure. Ducts of this system exhaust air from toilets, lockers, and storage rooms.

On the first floor, two-wall mounted fans exhaust air: one, in the kitchen, a 24 in. fan, with gravity type louvers, running at 500 r.p.m.; the other, in a toilet, at 500 c.f.m., a wall mounted fan with a 9½ in. shutter and frame outside. Small wall mountted fans are used also to ventilate the midway small building and the intermediate stations, in washrooms and pantries.

A feature of the piping from the Midway Main Building is the long run (150 ft.) of 1¼ in. lines through the tunnel under the Turnpike to the small midway building on the opposite side. These pipes are furred against the top of this tunnel and are well insulated.

The general contractor for the service buildings was The Austin Co. The heating was designed by the Construction and Maintenance Division, Standard Oil Co. of Pennsylvania, and installed by Snively Plumbing & Heating Co., Pittsburgh.

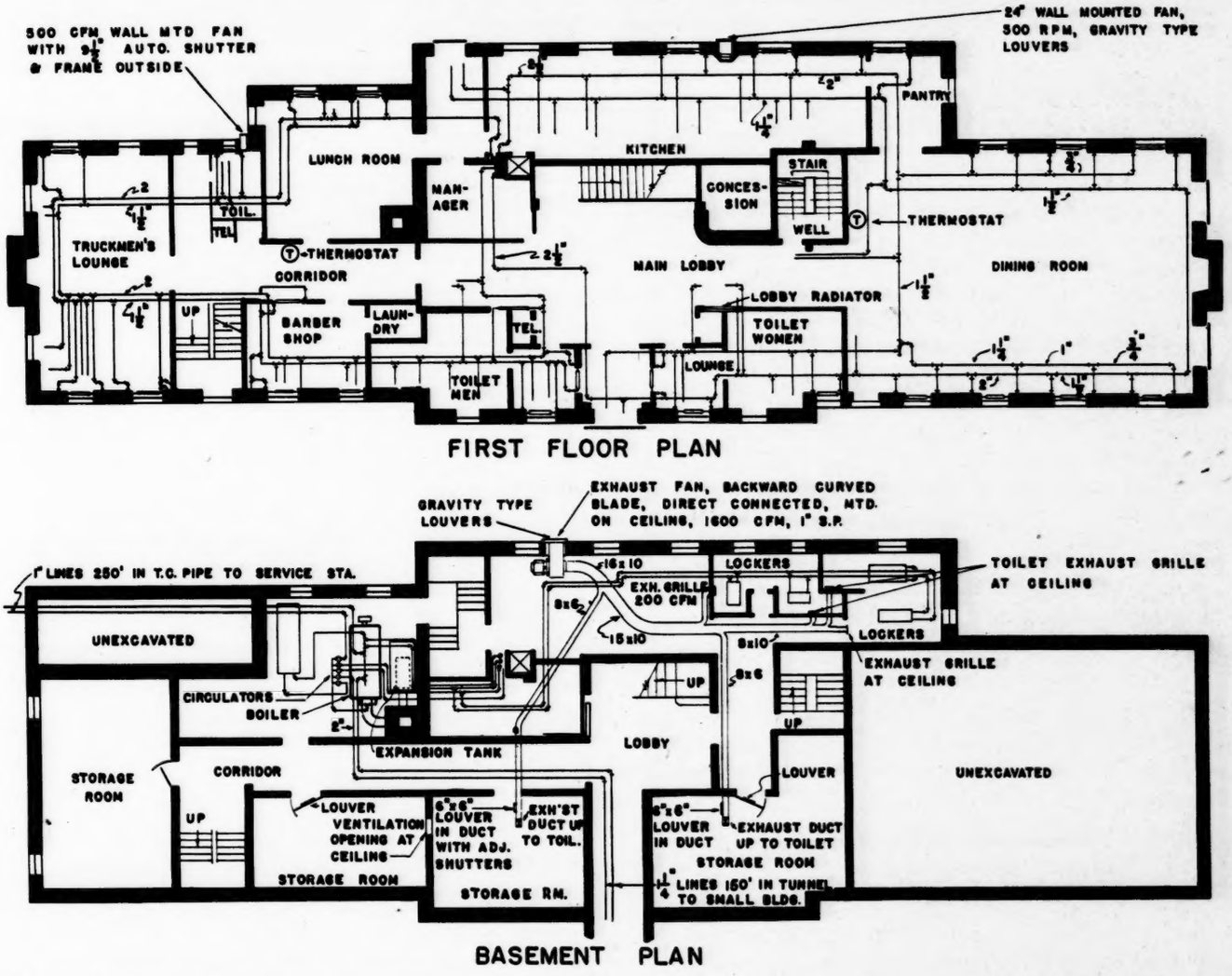
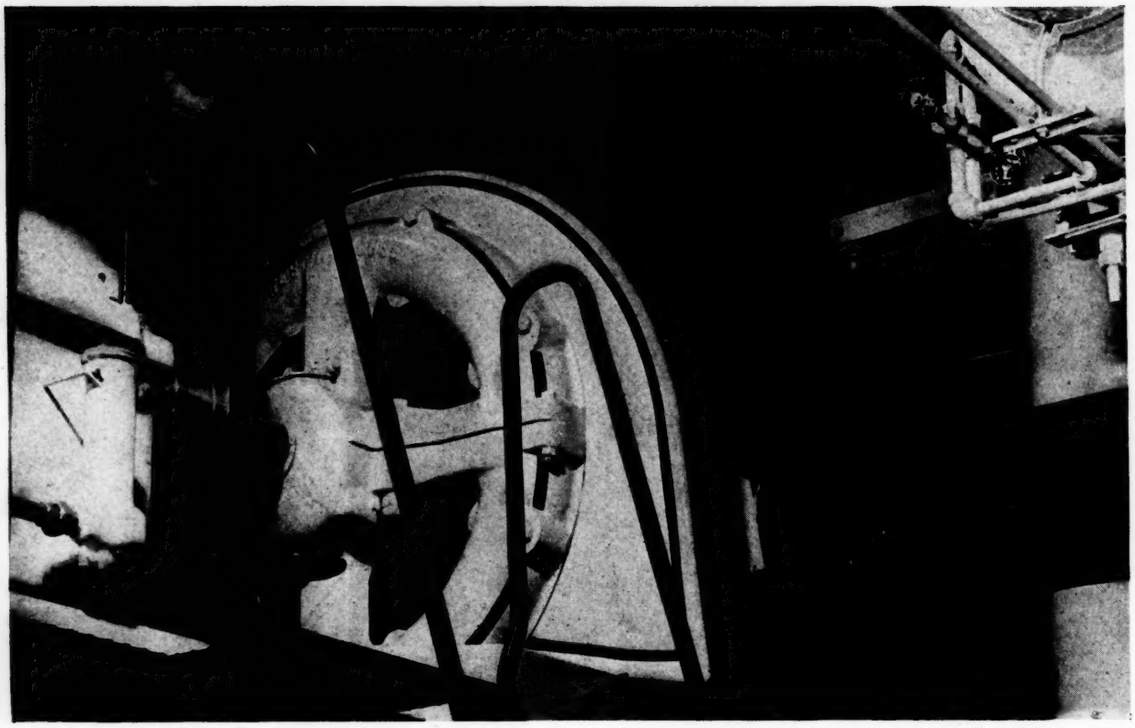


Fig. 9. Basement and first floor plan of the Midway Main Building on the Pennsylvania Turnpike, showing heating details. The circulators control the circulation of hot water to various independently controlled sections of this building and two dependent service stations. Basement ventilation is aided by the exhaust duct system shown in the diagram. First floor is exhausted by means of the two small mounted fans shown in the first floor plan. Two thermostats are shown on the first floor; another is installed on the second floor, plan of which is not shown here.



Maintenance and Care of Air Conditioning Equipment

By FRED D. MOSHER

THE statement that "proper maintenance pays dividends," becomes nothing more than a trite remark if the "proper maintenance" is not followed systematically. In spite of the fact that large scale air conditioning has entered its second decade of existence in the United States, there is a tendency to accept it as incidental to our economic life and not as the important factor that it is. From this trend of thought grows thinking that puts a literal interpretation on the word "automatic" in connection with air conditioning apparatus. Gross neglect results in failures and manufacturers are maligned for imaginary laxity. Brains and skilled hands, properly instructed, are as necessary for the functioning of air conditioning machines as are the gadgets that open and close dampers or valves; and it will remain that way until some unheard of genius develops the ultimate in mechanical men who will grab, for instance, a halide torch and quickly check a system for leaks.

Manufacturers actually spend thousands of dollars annually to acquaint users of their equipment with its nature and proper treatment. They recognize, also, that like modern medicine, good medicine is the preventive kind. Nobody desires air conditioning systems to operate satisfactorily more than the builders.

A prerequisite to the proper maintenance of any machine or group of machines is a thorough understanding of the functioning of such machines. It is not enough to know the "operation" of the equipment; the individual functions should be understood so that the correct diagnosis can be made when trouble occurs. It is also important that symptoms of trouble be readily recognized.

Next in importance to knowledge of the equipment is an understanding of the necessity for regular inspections—and the inspections should be systematic. Potential sources of trouble cannot escape detection where thorough inspection is practiced. So many failures are the results of progressive conditions that they can be practically eliminated by regular inspections. A case in point: A few years ago the writer was present at a post-mortem held on a condenser that had failed at the peak of the air conditioning season. The condenser supplied cooling for the refrigerant of a large compressor, the condensing water being re-circulated through a forced draft cooling tower on the roof of the building in which the equipment was housed. Acidity of the water corroded the condenser tubes until they eventually failed. In this instance the condition of the water had never been checked until *after* the failure. The emergency repairs cost a large sum of money and a prolonged outage; it was an extremely expensive lesson.

AIR CONDITIONING MAINTENANCE SCHEDULE

CENTRAL SYSTEMS

REFRIGERANT CIRCUIT CONTROLS—the part of the system through which the refrigerant circulates including the compressor, connected piping and valves, condenser receiver, and the evaporator. The controls consist of such devices as thermostatic expansion valves, switches and regulators with automatic cut-outs and solenoid valves for water and refrigerant.

DAILY—System should be inspected for overheating of the moving parts, "knocking," excessive pressures in the system of piping and vessels. The proper cutting in and cutting out points of machines should be noted. Any evidences of sticking of the expansion valves should be noted since such sticking is usually caused by moisture in the system. Any signs of leakage should be traced to their source.

MONTHLY—On monthly inspections the refrigerant circuit should be checked for leaks of the refrigerant. The standard halide refrigerant gas leak detector is recommended for this purpose. Also look for oil leaks around packings and water leaks throughout the system. Scale traps should be checked. Purge valve should be opened or the system should be purged through the means provided. Check thermostatic gas-filled bulbs for tightness against pipes and moisture between the bulb and the pipe; moisture may freeze under certain conditions and break the contact necessary for good operation of such devices. Remove the covers from all pressure switches and such control mechanisms at least once a month as loose screws, springs, and contacts will cause trouble if not checked in time.

YEARLY—Safety devices such as relief valves should be removed and tested unless it is required more frequently by local ordinances. Valves suspected of leaking should be examined for wear. Packings should be inspected. It is assumed that at the time of the yearly inspection the system is out of service so that parts normally inaccessible can be dismantled to check for wear. A summation of the year's record should be made at this time including the notes made of the yearly inspection.

CONDENSING CIRCUIT—the part of the system through which the condenser cooling water circulates and consists of condensers, water piping, floats, valves; if the water is recirculated it may include spray or cooling ponds, cooling towers, settling tanks, and collection pans; or it may include an evaporative type condenser.

DAILY—The condition of the water should be checked for corrosive tendencies. Temperatures should be taken daily since they often are indications of the cleanliness of the system. Quantity of makeup should be recorded daily when means are at hand to do so. Float valves should be inspected daily since improper functioning of floats may result in waste of water. Leaks, even though not serious, should be noted.

MONTHLY—Sumps, tanks, and collection pans should be checked for slime formation and algae growth, and for general cleanliness. Water samples should be taken for chemical analysis. Spray nozzles should be inspected for plugging. Accessible parts of evaporative type condensers should be given a thorough inspection each month for accumulations in the air and water passages.

YEARLY—Water should be drained from all parts of the system so that all internal parts can be properly inspected. The circulating system should be checked for clogging and partial stoppages. Any valves suspected of leaking during operation should be dismantled for inspection at this time. All internal surfaces should be carefully scrutinized for corrosive action and scale formation. There should be a general summing up of the yearly record at the time of annual inspections. All items carried over from the other inspections will now become a part of the yearly report.

COOLING WATER SYSTEM—the circuit through which the refrigerated water passes. Includes storage tanks, evaporators, piping, spray units or washers, and collection pans and heating coils.

DAILY—The storage tank should be checked for water level and condition. Floats should be inspected daily. On new systems the spray nozzles in the units should be inspected daily for plugging as small particles of dirt find their way to the nozzles for the first month or so of operation.

MONTHLY—The storage tank should be drained monthly for inspection; the large amount of dirt washed out of the air may cause plugging if a careful check is not made. After daily inspection of the spray nozzles has been dropped it should be made monthly. A water sample should be taken for chemical analysis. Drains should be inspected.

YEARLY—Water should be drained from the cooling water circuit for the annual inspection. This inspection should include a check of evidences of scale formation and corrosion.

FILTERS AND DUCTS—the part of the system which cleans and carries the air of the system. Because of their structural nature they require less attention than other parts of the system.

DAILY—Static pressure drop across filters should be checked.

MONTHLY—Filters should be inspected for dust accumulations. Sections should be removed at random from sectional filters each for checking. Insulated duct work should be inspected monthly for traces of moisture getting between insulation and duct walls.

YEARLY—The general condition of filters should be determined. Ducts are inspected at this time for defects and leakages. Dirt accumulations are noted, and any abnormal conditions investigated.

ROTATING APPARATUS—those parts of the air conditioning system, such as pumps, compressors, blowers, motors, and the controlling devices which convert or assist in the conversion of electrical energy into refrigeration.

DAILY—Motors should be given a daily "once over" with particular attention being given to automatic starters and contactors. Any arcing should be noted. Such equipment properly cared for seldom gives trouble.

MONTHLY—Note the condition of brushes, protective devices and oil levels. Belt tensions should be checked and any apparent misalignment or abnormal wear should be fully investigated. Blower bearings should be checked for proper adjustment. Pump packings should be inspected monthly for leakage. The casing covers of motors used with induced draft cooling towers should be removed once a month for inspection. Where gears are used on this type fan they should be inspected once a month.

YEARLY—In its turn each piece of equipment should be taken out of service for a thorough annual inspection. Clearances should be checked on reciprocating parts. Wear of bearings should be checked. Evidences of stress should be noted and possible cracks should be investigated. Each lubricating system should be examined for dirt and grit. Important safety devices are tested at this time. A report should be summarized on the general condition of all the rotating machines, the drives and the driven units.

UNIT AIR CONDITIONERS

UNIT AIR CONDITIONERS—self-contained air conditioning systems that are built in compact units.

DAILY—Since these units are as near "fool proof" as it is possible to make them, inspection becomes co-incidental with "operation."

MONTHLY—The system should be checked for leaks with a standard halide gas leak detector. Belts should be inspected each month for signs of abnormal wear or pulley misalignment. The builder's instructions on purging should be followed. Oil levels are checked each month. Knocks should be investigated.

YEARLY—A check should be made on the water being used to determine its condition. The service of the small unit does not warrant other than reasonable precautions to see that clean water is at all times supplied to the system. At the annual inspection the water in the system should be drained and the various parts examined for corrosion and scale. Safety devices should be removed and checked and their condition noted. An over-all "condition" record should be made for the year. In some cases it is recommended that a general check-up or inspection be made of unit systems after a stipulated number of service hours; quite often the number of service hours is much longer than a year. The recommendations based on service hours should always be followed because the maintenance requirements are a function of hours of actual operation which determines the life of individual parts of the unit.

This example is probably typical of many failures, all due to the same cause—neglect.

It is assumed that some form of daily log is kept to record hours of operation and consumption of water and oil and other pertinent data. Systematic inspections also require that records be kept. Such records should show the condition of apparatus at the time of inspection and any repairs that were found necessary and subsequently completed. These records need not be elaborate to the point of what some may call "bookkeeping," but they should carry ample data to reveal any short life in connection with individual parts or any abnormal costs being run up for maintenance.

In general, inspections should be divided into daily, monthly and yearly inspections. Daily inspections are those that have to do with checking water levels, water conditions, heating up of parts, what little hand oiling that is necessary, or what may be termed a daily "once over." Monthly inspections are usually those specified by the builders as routine checks on oil levels in reservoirs, checks for leaks of the refrigerant, oil, and water; cleaning of electrical apparatus, filters, spray nozzles, collection pans; checking of control devices, belt tensions, scale traps, and purging. Yearly inspections require the opening up of condensers for examination, dismantling of machines, and any inspections requiring the apparatus to be shut down for prolonged periods. For inspection and maintenance purposes it is desirable to group the apparatus under their logical headings. The accompanying table of maintenance schedules is intended to serve as a guide for the proper up-keep of a typical air conditioning system, and the grouping has been arranged in what is considered a logical order for efficient work.

It might appear that the schedules outlined above are an attempt to reduce inspection to a rigid routine. No such intention was in mind. Quite the contrary, the grouping was definitely arranged to permit flexibility for physical reasons unique to any given plant. Most failures of equipment are, without question, preventable. A pulley that causes frequent replacements of belts is being neglected by somebody; and the relief valve that fails to open the one time it is required to, has been a forgotten piece of equipment for some time. Even in an extreme case it may take a dollar's worth of maintenance to prevent a dollar repair; the maintenance dollar, however, is the wiser one because the failure that necessitated the repair may be accompanied by intangibles—not easily evaluated in terms of dollars and cents.

How to Determine Size and Cost of Freon Lines

By WILLIAM PARKERSON

Of considerable importance to air conditioning engineers is the problem of sizing freon refrigerant lines. In spite of this much of the previous information and data which has been published on this subject is either lacking in completeness or is difficult to use. In this article, which is to be published in three parts, the author develops a practical set of curves for correctly sizing freon lines, for determining the velocity of freon travel and for estimating the cost of materials in the lines. In this first part the author develops the formula and presents tabular data needed for designing refrigerant lines.

MUCH has been written and spoken regarding the design of refrigerant lines. However, this subject has not enjoyed the attention it deserves, in either the magazines or the texts of the industry. Moreover, what curves, charts, and tables are available seem, to the writer, to be either lacking in completeness or clumsy to use. For instance, some are said to include "an average number of fittings"; some include a certain percentage allowance for elbows and fittings; some are based on 90F condensing temperature (seldom used in practice); few give any idea of velocity and others are based on velocity alone; still others read inconveniently, in pounds per square inch per hundred feet; and so on. In attempting to meet, answer and overcome the above faults or shortcomings, the present paper assumes a threefold purpose; namely, to develop a rational and conveniently usable set of curves for 1, the proper sizing of freon lines; 2, the velocity of fluid travel through the pipes; and 3, the probable cost of the materials involved in the construction of the said lines.

Part I — Sizing Freon Lines

It is a matter of common knowledge that the more thoroughly one understands the principles underlying any table, chart, or curve, the more intelligently, and consequently the more successfully, one can apply the information presented therein. For this reason, it was considered worthwhile to inquire into the fundamentals or rudiments of refrigerant line design and attempt an understandable presentation of the practical side of the whole subject, and in so doing, to devise a usable set of tables and curves containing all the necessary design information. To this end, let us first develop the general equation for refrigerant line design, commencing with fundamentals. Everyone is familiar with the so-called Darcy formula:

$$h_f = \frac{f_1 L v^2}{D2g} \quad \ldots\ldots (1)$$

Now $$P = \frac{h_f W}{144} = \frac{W}{144} \times \frac{f_1 L v^2}{D2g} \quad \ldots\ldots (2)$$

But

$$v = \frac{\text{Cu. Ft. Per Sec.}}{A} = \frac{M}{W \times 60 \times A} = \frac{M}{60 \times W \times \frac{\pi D^2}{4}}$$

$$= \frac{4M}{60\pi WD^2} \quad \text{or} \quad v = \frac{M}{15\pi WD^2} \quad \ldots\ldots (3)$$

Substituting (3) in (2), we have,

$$P = \frac{W}{144} \times \frac{f_1 L}{D2g} \times \left(\frac{M}{15\pi WD^2}\right)^2$$

$$P = \frac{W}{144} \times \frac{f_1 L}{2 \times 32.2 \times D} \times \frac{M^2}{(15\pi)^2 \times W^2 \times D^4}$$

$$P = \frac{f_1}{144 \times 2 \times 32.2 \times 15 \times 15 \times \pi^2} \times \frac{M^2 L}{WD^5}$$

But substituting $D = d/12$

$$P = \frac{(12)^5 f_1}{144 \times 2 \times 32.2 \times 15 \times 15 \times \pi^2} \times \frac{M^2 L}{Wd^5}$$

$$P = 0.0121 f_1 \frac{M^2 L}{Wd^5} \quad \ldots\ldots (4)$$

Symbols

A = Inside area of pipe in square feet.
d = Inside diameter of pipe in inches.
D = Inside diameter of pipe in feet.
f = Coefficient of friction.
g = Acceleration of gravity = 32.2 feet per second per second.
h_f = Head lost in fluid friction in feet.
H_f = Heat content of liquid refrigerant in B.t.u. per pound.
H_g = Heat content of refrigerant vapor in B.t.u. per pound.
H = Refrigerating effect in B.t.u. per pound.
L = Equivalent length of pipe line in feet.
L_m = Measured length of straight pipe in feet.
L_e = Feet of straight pipe equivalent to n elbows.
M = Pounds of liquid or vapor per minute transmitted through line.
n = Number of 90 degree elbows in line.
P = Pressure drop in pounds per square inch corresponding to h_f.
P_c = Conversion factor in pounds per square inch per degree Fahrenheit.
S = Velocity in feet per minute.
T = Tons of refrigeration.
Δt = Saturated-vapor temperature-equivalent of "P".
v = Velocity in feet per second.
V = Specific volume in cubic feet per pound.
W = Density in pounds per cubic foot.
X = (See table II.)
Z = Viscosity in centipoises. (Tables IV, V, VI.)

Note:—"P_c" may also be thought of as the saturated-vapor pressure-equivalent of one degree Fahrenheit change in temperature.

It is interesting to note, in passing, that, for the flow of steam in pipes, experimentation demonstrated that $0.0121f_1$ could be replaced by the quantity $0.0001321\ (1 + 3.6/d)$ and thus came into being the Babcock formula:

$$P = 0.0001321\left(1 + \frac{3.6}{d}\right)\frac{M^2L}{Wd^5}$$

Now

$$M = \text{Lbs./Min.} = \text{B.t.u./Min.} \div \text{B.t.u./Lb.} = 200T \div H$$

$$M = \frac{200T}{H} \quad \ldots\ldots (5)$$

Also $$W = 1/V \quad \ldots\ldots (6)$$

Substituting (5) and (6) in (4), we have,

$$P = \frac{0.0121f_1\,(200T)^2\,LV}{H^2d^5} = \frac{0.0121 \times 40{,}000f_1T^2VL}{H^2d^5}$$

$$P = 484f_1\frac{T^2VL}{H^2d^5} \quad \ldots\ldots (7)$$

Here, the task of evaluating the friction factor, f_1, presents itself.

Now on page 126 of the fourth edition of "Fan Engineering," may be found a formula which would appear as follows when written in the symbols of this paper:

$$P = \frac{0.433f_2L\,(n \div 1488)^a W^b v^x}{(d \div 12)^y} \quad \ldots\ldots (8)$$

Here, also, are given the following values for copper tubing:

$$f_2 = 0.00005,\ a = 0.2,\ b = 0.8,\ x = 1.8,\ y = 1.2 \quad \ldots\ldots (9)$$

Now substituting (3) and (9) in (8) and remembering that

$$D = d/12 \text{ and } W = 1/V$$

equation (8) may be replaced to:

$$P = 10.22\frac{n^{0.2}T^{1.8}VI}{H^{1.8}d^{4.8}} \quad \ldots\ldots (10)$$

or $$P = 10.22\left(\frac{nHd}{T}\right)^{0.2} \times \frac{T^2VL}{H^2d^5} \quad \ldots\ldots (11)$$

In order for equations (11) and (7) to be identical, we must have,

$$484f_1 = 10.22\left(\frac{nHd}{T}\right)^{0.2}$$

or $$f_1 = 0.0211\left(\frac{nHd}{T}\right)^{0.2} \quad \ldots\ldots (12)$$

Thus, we would seem to have arrived at a value for the friction factor f_1. But let us search further for a confirmation or ratification of equation (11), in order to justify our conclusion.

In the splendid article of Gygax and Willson, entitled "Pressure Drop in Refrigerant Liquid Lines," appearing in the February, 1940 issue of Refrigerating Engineering, may be found a formula which, when rewritten in the symbols of this paper, appears thus:

$$P = \frac{1940f_3T^2VL}{H^2d^5} \quad \ldots\ldots (13)$$

In order for equations (13) and (11) to be identical, we must have,

$$1940f_3 = 10.22\left(\frac{nHd}{T}\right)^{0.2}$$

or $$f_3 = 0.00527\left(\frac{nHd}{T}\right)^{0.2} \quad \ldots\ldots (14)$$

In the above article, f_3 is stated to be some function of F, which is defined as follows:

$$F = \frac{9.793T}{nHd} \quad \ldots\ldots (15)$$

From this it is clear that f_3 must be some function of $9.793/F$, for which we may obviously write:

$$\frac{9.793}{F} = \frac{nHd}{T}$$

The following numerical relations between F and f_3, are given in the article:

F	f_3
0.27- 0.33	0.0125
0.59- 0.68	0.0095
1.15- 1.20	0.0082
1.80- 1.89	0.0072
3.41- 3.55	0.0062
6.11- 6.50	0.0053
15.1- 17.5	0.0043
52.0- 70.0	0.0033
121.0- 140.	0.0028

Now using the average values for F, we have:

F	f_3	$\frac{9.793}{F} = \frac{nHd}{T}$
0.30	0.0125	32.6
0.635	0.0095	15.4
1.175	0.0082	8.33
1.845	0.0072	5.30
3.480	0.0062	2.81
6.305	0.0053	1.55
16.30	0.0043	0.600
61.00	0.0033	0.1604
130.5	0.0028	0.075

By plotting the values of f_3 against the values of $\frac{\pi Hd}{T}$ on log-log paper, we find that,

$$f_3 = 0.005\left(\frac{nHd}{T}\right)^{0.23} \quad \ldots\ldots (16)$$

The close similarity of equations (16) and (14) is viewed as a confirmation of equation (11).

This, then, allows us to sum up as follows:

1. We have accomplished the task of finding a reliable value of the friction factor f_1. See equation (12).
2. We may, for the sake of completeness, rewrite the equation of Gygax and Willson thus:

$$P = 1940\left[0.005\left(\frac{nHd}{T}\right)^{0.23}\right]\frac{T^2VL}{H^2d^5}$$

$$= 9.7\left(\frac{nHd}{T}\right)^{0.23}\frac{T^2VL}{H^2d^5} \quad \ldots\ldots (17)$$

or $$P = 9.7\frac{n^{0.23}T^{1.77}VL}{H^{1.77}d^{4.77}} \quad \ldots\ldots (18)$$

3. We may select, as the basis for further calculation, equation (10):

$$P = 10.22\frac{n^{0.2}T^{1.8}VL}{H^{1.8}d^{4.8}}$$

Since, in refrigeration and air-conditioning design practice, the engineer deals more directly with temperatures than with pressures, the curves to be developed and presented herein will be more usable if we employ Δt (the saturated-vapor temperature-equivalent of P) rather than P. Therefore, let us substitute $P = P_c\,\Delta t$ in equation (10),

$$P = P_c\,\Delta t = 10.22\frac{n^{0.2}T^{1.8}VL}{H^{1.8}d^{4.8}}$$

TABLE I.—PHYSICAL PROPERTIES OF FREON

Temp. F. t	Specific Volume In Cu. Ft. Per Lb.		Heat Content In Bt.u. Per Lb.		Viscosity In Centipoises		Conversion Factor or Pressure-Equivalent Lbs. Sq. In. Degree P_c
	Liq. V	Vap. V	Liq. H_f	Vap. H_g	Liq. π	Vap. π	
30	0.0115	0.939	14.76	81.61	0.296	0.0118	0.795
40	0.0116	0.792	17.00	82.71	0.286	0.0119	0.907
50	0.0118	0.673	19.27	83.78	0.277	0.0121	1.035
95	0.0126	0.342	29.93	88.19	0.245	0.0128	1.750
105	0.0128	0.297	32.40	89.03	0.239	0.0129	1.900
115	0.0131	0.258	34.90	89.80	0.234	0.0131	2.100

TABLE II.—GOVERNMENT TYPE "L" COPPER TUBING

Line Size In Inches O.D.	Inside Area In Sq. Ft. A	Inside Diam. In Inches d	15d	$10.222/d^{4.8}$ X
3/8	0.00054	0.315	4.725	2628.
5/8	0.0016	0.545	8.175	188.3
7/8	0.0034	0.785	11.775	32.67
1 1/8	0.0057	1.025	15.375	9.080
1 3/8	0.0087	1.265	18.975	3.308
1 5/8	0.0124	1.505	22.575	1.437
2 1/8	0.0214	1.985	29.775	0.3805
2 5/8	0.0332	2.465	36.975	0.1346
3 1/8	0.0474	2.945	44.175	0.0573
3 5/8	0.0640	3.425	51.375	0.0278
4 1/8	0.0830	3.905	58.575	0.0148
5 1/8	0.1295	4.875	73.125	0.0051

TABLE III.—REFRIGERATING EFFECT OF FREON

Suct. Temp. F	Cond. Temp. F	Heat Content		Refrigerating Effect H	$H^{1.8}$
		B.t.u. Per Lb. H_g	B.t.u. Per Lb. H_f		
30	95	81.61	29.93	51.68	1213.3
30	105	81.61	32.40	49.21	1110.9
30	115	81.61	34.90	46.71	1011.4
40	95	82.71	29.93	52.78	1260.2
40	105	82.71	32.40	50.31	1156.0
40	115	82.71	34.90	47.81	1054.7
50	95	83.78	29.93	53.85	1306.6
50	105	83.78	32.40	51.38	1200.7
50	115	83.78	34.90	48.88	1097.6

From which:

$$\Delta t = 10.22 \frac{n^{0.2} T^{1.8} V L}{H^{1.8} d^{4.8} P_c} \quad \ldots\ldots\ldots\ldots (19)$$

This is the general equation underlying all of our families of curves.

Now, for the sake of convenience, let

$$X = \frac{10.22}{d^{4.8}} \text{ and } Z = \frac{n^{0.2} V}{H^{1.8} P}$$

then,

$$\Delta t = XZLT^{1.8} \quad \ldots\ldots\ldots\ldots (20)$$

Table I presents the physical properties of freon, pertinent to this discussion; Table II, the necessary data on Government Type L copper tubing and the values of "X" resulting therefrom. Table III shows the refrigerating effect for nine selected sets of conditions. Tables IV, V, and VI give the values of "Z" for any set of conditions within the scope of this paper.

Using equation (20) and Tables II, IV, V, VI, and VIII, it becomes a relatively simple matter to determine what length of a given size pipe will transmit a given tonnage with a given friction drop in the line, under selected conditions. For example, suppose we wish to know through what equivalent length of 2⅝ in. O.D. suction line we may transmit 30 tons of refrigeration with a 2F friction drop; suction temperature being 40F and condensing temperature, 105F. From Tables II and IV we find

$$X = 0.1346 \quad \text{and} \quad Z = 0.00031134$$

and from Table VIII,

$$T^{1.8} = 455.85$$

Substituting these values in equation (20), we have

$$2 = 0.1346 \times 0.00031134 \times L \times 455.85$$

or

$$L = 2 \div (0.1346 \times 0.00031134 \times 455.85) = 104.6 \text{ feet.}$$

Now if we rearrange equation (20) thus:

$$T = \sqrt[1.8]{\frac{\Delta t}{XZL}} \quad \ldots\ldots\ldots\ldots (21)$$

we see that the tonnage that may be

TABLE IV.—SUCTION LINE DESIGN DATA

Suct. Temp. F	Cond. Temp. F	$\pi^{.2}$	Sp. Vol. Cu. Ft. Per Lb. V	Press. Equiv. P_c	$\frac{\pi^{.2} V}{H^{1.8} P_c}$ Z	% Variation From Mean Value of Z, Taken Two at a Time	
						Varying Suct. Temp. Only	Varying Cond. Temp. Only
30	95	0.4113	0.939	0.795	0.00040039		
30	105	0.4113	0.939	0.795	0.00043729		4.40
30	115	0.4113	0.939	0.795	0.00048031		4.69
40	95	0.4124	0.792	0.907	0.00028560	16.7	
40	105	0.4124	0.792	0.907	0.00031134	16.8	4.31
40	115	0.4124	0.792	0.907	0.00034125	16.9	4.58
50	95	0.4135	0.673	1.035	0.00020579	16.2	
50	105	0.4135	0.673	1.035	0.00022394	16.3	4.23
50	115	0.4135	0.673	1.035	0.00024498	16.4	4.48
Average % Variation from Mean Value of Z						16.55	4.45

TABLE V.—LIQUID LINE DESIGN DATA

Suct. Temp. F	Cond. Temp. F	$\pi^{.2}$	Sp. Vol. Cu. Ft. Per Lb. V	Press. Equiv. P_c	$\frac{\pi^{.2} V}{H^{1.8} P_c}$ Z	% Variation From Mean Value of Z, Taken Two at a Time	
						Varying Suct. Temp. Only	Varying Cond. Temp. Only
30	95	0.7549	0.0126	1.75	0.0000044794		
30	105	0.7513	0.0128	1.90	0.0000045561		0.85
30	115	0.7479	0.0131	2.10	0.0000046128		0.62
40	95	0.7549	0.0126	1.75	0.0000043127	1.90	
40	105	0.7513	0.0128	1.90	0.0000043783	1.99	0.75
40	115	0.7479	0.0131	2.10	0.0000044234	2.10	0.51
50	95	0.7549	0.0126	1.75	0.0000041597	1.80	
50	105	0.7513	0.0128	1.90	0.0000042156	1.89	0.67
50	115	0.7479	0.0131	2.10	0.0000042507	1.99	0.41
Average % Variation from Mean Value of Z						1.95	0.64

transmitted through a line of given length and size with a given friction drop is inversely proportional to the 1.8 root of "Z." Therefore, inquiry into the variations of "Z" with changing refrigerant conditions will show how line capacities change with refrigerant conditions, remembering that, as "Z" increases, line capacity decreases, and vice versa.

This inquiry may be conducted as follows: Refrigerant conditions are listed in Tables IV, V, and VI in steps of 10F, both suction temperature and condensing temperature. It will be noticed that any change in either the suction temperature or the condensing temperature changes the value of "Z." A measure of the sensitiveness of "Z" to changes in refrigerant conditions is given in the last two columns of Tables IV, V, and VI. It is the percent variation from the mean value of "Z," taking two values at a time, changing one temperature 10F, while holding the other constant. Thus, in Table IV, when we hold the suction at 30F and increase the condensing temperature from 95F to 105F, the variation from the mean value of "Z" is

$$\frac{0.00043729 - 0.00040039}{0.00043729 + 0.00040039} \times 100 = 4.40$$

Likewise, when we hold the condensing temperature at 95F and increase the suction from 30F to 40F, the per cent variation of "Z" from its mean value is

$$\frac{0.00040039 - 0.00028560}{0.00040039 + 0.00028560} \times 100 = 16.7$$

From this, it is evident that the capacity of the suction line is very sensitive to changes in suction temperature, but only mildly sensitive to changes in condensing temperature.

From Table V, it may be seen that liquid line capacity is almost independent of variations in suction temperature and/or condensing temperature.

Similarly, it may be seen from Table VI that hot-gas line capacity is almost independent of suction temperature changes, but is somewhat sensitive to variations in condensing temperature.

The above considerations lead us to the decision to construct six families of curves, namely, three for the underlined set of conditions in Table IV, one for the underlined set of conditions in Table V, and two for the underlined sets of conditions in Table VI.

It is interesting to note in passing that an increase in condensing temperature decreases the capacity of suction and liquid lines and increases the capacity of the hot-gas line; while on the other hand, an increase in suction temperature increases the capacity of all three lines.

Table VII lists the friction of 90° standard elbows in equivalent length of straight pipe and in diameters. These values were taken from four sources selected almost at random. Source 3 is a common formula found in a good many of the older text-books. Source 4 is of special interest on account of the accompanying statement that the data presented here are based upon actual tests of valves and fittings. The average of all diameter multipliers in the table is 30.12. Therefore, while no great accuracy is claimed for the figure, 30 diameters seems a conservative and consistent relationship to employ herein. How it compares with other values is shown by the last column in Table VII.

Now that we have arrived at the resistance value of an elbow, we must consider the proper number of elbows to include in a line. In other words, we must decide for ourselves how many is *an average number*. With this in mind, a large number of actual installations were studied. With surprising regularity, six

TABLE VI.—HOT GAS LINE DESIGN DATA

Suct. Temp. F	Cond. Temp. F	π^{-2}	Sp. Vol. Cu. Ft. Per Lb. V	Press. Equiv. P_e	$\frac{\pi^{-2}V}{H^{1.8}P_e}$ Z	% Variation From Mean Value of Z, Taken Two at a Time: Varying Suct. Temp. Only	Varying Cond. Temp. Only
30	95	0.4182	0.342	1.75	0.000067515		
30	105	0.4192	0.297	1.90	0.000058981		6.75
30	115	0.4201	0.258	2.10	0.000051033		7.23
40	95	0.4182	0.342	1.75	0.000065003	1.89	
40	105	0.4192	0.297	1.90	0.000056680	1.99	6.85
40	115	0.4201	0.258	2.10	0.000048939	2.09	7.20
50	95	0.4182	0.342	1.75	0.000062694	1.81	
50	105	0.4192	0.297	1.90	0.000054573	1.90	6.92
50	115	0.4201	0.258	2.10	0.000047028	1.99	7.42
Average % Variation from Mean Value of Z						1.95	7.06

TABLE VII.—FRICTION OF 90° STANDARD ELBOWS. EQUIVALENT LENGTH OF STRAIGHT PIPE IN FEET AND IN DIAMETERS

Nominal Size of Pipe In Inches		Source 1	Source 2	Source 3	Source 4	30d+12
3/4	Feet	2.2	2.6	0.8	1.3	1.9
	Diameters	36.2	41.6	13.3	20.8	30.0
1	Feet	2.8	3.4	1.4	1.8	2.5
	Diameters	33.6	40.3	16.5	21.8	30.0
1¼	Feet	3.8	4.4	2.0	2.2	3.1
	Diameters	36.5	42.0	19.6	21.1	30.0
1½	Feet	4.4	5.1	2.8	2.7	3.8
	Diameters	35.2	40.8	22.3	21.6	30.0
2	Feet	5.6	6.5	4.5	3.6	5.0
	Diameters	33.6	38.8	27.2	21.6	30.0
2½	Feet	6.7	7.7	6.5	4.4	6.3
	Diameters	32.2	37.1	31.1	21.1	30.0
3	Feet	8.2	9.4	8.6	5.3	7.5
	Diameters	32.8	37.6	34.5	21.2	30.0
3½	Feet	9.5	....	10.9	6.2	8.8
	Diameters	32.6	....	37.4	21.3	30.0
4	Feet	11.0	12.3	13.3	7.0	10.0
	Diameters	33.0	36.9	39.9	21.0	30.0
Average Diameters.		34.0	39.4	26.8	21.3	30.0

Source: No. 1—"Trane Air-Conditioning Manual." First Edition. Page 319.
Source: No. 2—"Fan Engineering." Fourth Edition. Page 147.
Source: No. 3—Equivalent Length = 6.33d ÷ (1 + 3.6/d).
Source: No. 4—"Walworth Company Catalogue No. 89." Page 502.

TABLE VIII.—VALUES OF $T^{1.8}$ IN TERMS OF T.

T	1	3	6	10	30	60	100	300
$T^{1.8}$	1	7.2247	25.158	63.096	455.85	1587.3	3981.1	28762

elbows turned out to be the average number per line and this average was found to be largely unaffected by the size of the job or the distance separation between units. This fact would appear to discredit any *percentage allowance for elbows*. While the average of six elbows per line has been adopted for the purpose of this discussion, curves have been prepared in such a manner as to allow the user thereof to apply simple corrections, as will be demonstrated later.

Obviously, from the foregoing,

$$L_e = \frac{n \times 30d}{12}$$

and the length of straight pipe equivalent to six elbows is,

$$L_e = \frac{6 \times 30d}{12} = 15d$$

Values of 15d are given in Table II and these are the figures shown on the curves near the line sizes, as allowances for elbows.

(Next month the author will present six charts which make possible the quick and easy selection of proper size refrigerant lines, the determination of freon velocities and the cost of the lines.)

Minimum Setting Heights for Stokers

EDITOR, HEATING & VENTILATING:

A few months ago you ran an editorial commenting on the Minimum Setting Heights standards established by our Association. We appreciate your editorial comment and the compliments you paid our Association in this work. However, I desire to call to your attention a few statements made in the editorial which are either incorrect or misinterpreted by you.

You stated, "When fully substantiated by field experience, such a standard should prove most useful in meeting a problem which has some public interest generally because of its close relation to smoke prevention." In our press release dated April 16, 1940, we said: "These standards are based on the practical experience of the leading manufacturers, as well as dealers and installers, in the industry over a period of many years covering hundreds of thousands of installations plus research work which has been conducted by the Association in cooperation with national organizations in the . . ." This is the fact. These minimum setting heights are based preponderantly on the experience of our members on the installation of hundreds of thousands of stokers in different kinds and sizes of boilers under many different conditions, so that we wish to correct this statement I first quoted above which appeared in your editorial that ". . . when fully substantiated by field experience. . . ." Naturally, future field experience will have a bearing on practices and any revisions or standards which may be published in the future.

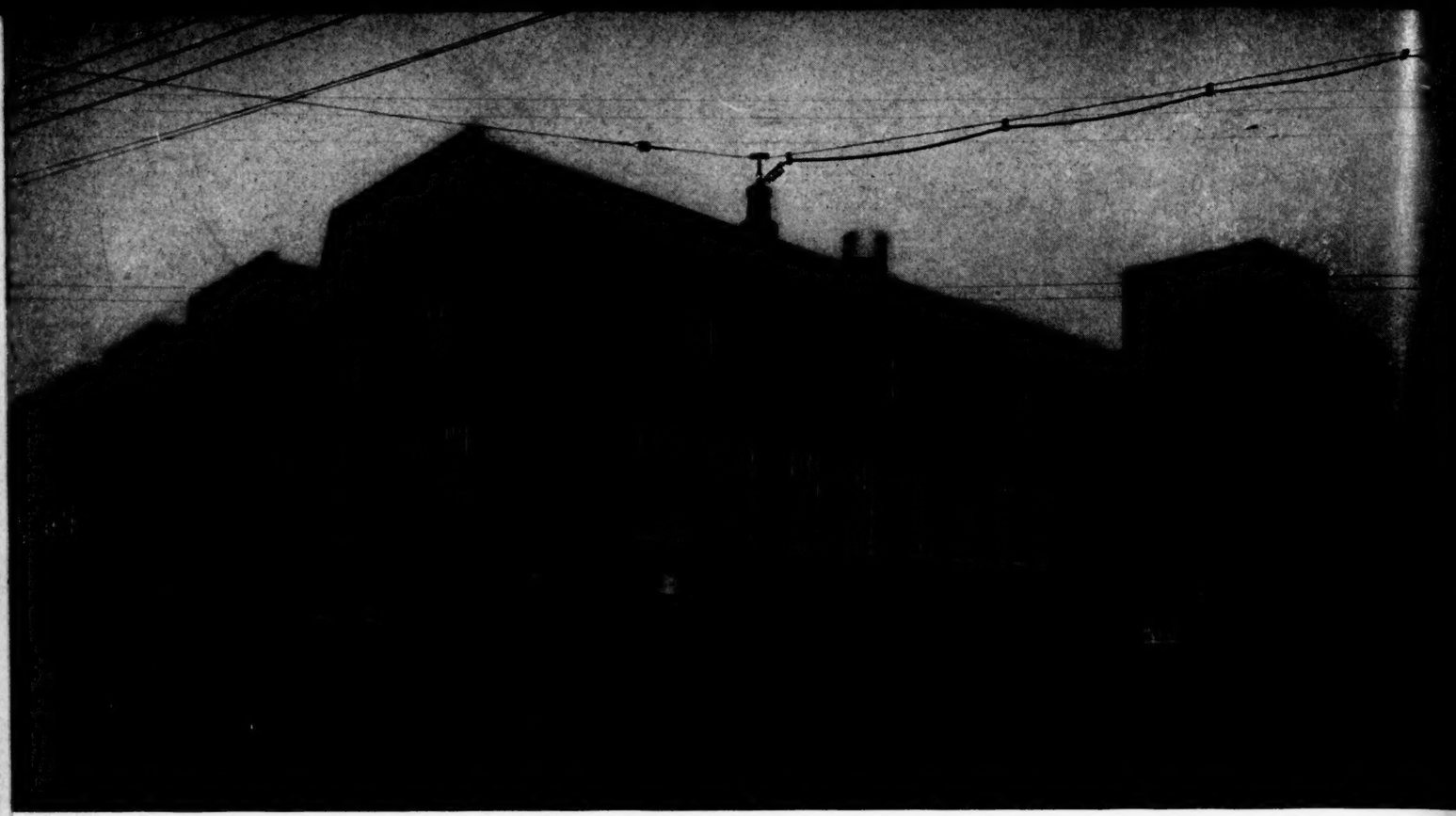
A statement was also made in your editorial, "This fixes the setting height at 27 inches at 100 pounds per hour." This statement infers that these setting heights are now all standardized and fixed. This is not the case. The code, as published, reads, "Recommended *Minimum* Setting Heights." The curve which shows, for example, a stoker setting at 27 inches for a hundred-pound machine is merely the *minimum*. There will be higher settings required in many cases and in some instances, under certain conditions, a stoker can actually be set lower than this and provide efficient and smokeless operation. These setting heights which have been established by our Association are merely *average* recommended *minimums*. It is still necessary for the stoker installer to analyze the individual requirements of a particular installation and adjust the setting heights accordingly in order to obtain the most satisfactory performance as there are a number of factors which often make it desirable to vary the setting height above or below the average recommended minimum as established in this standard. As a matter of fact the majority of stoker dealers and installers follow the individual recommendations of their manufacturer but the standards as established by our Association represent the average recommendations of our members but only as to allowable minimums, but again these minimums are merely set up as a guide and are deviated from according to requirements in every individual installation.

I feel that you would like to have our comment on your editorial, not in the spirit of criticism but so that you may be informed of the intent and purpose of these standards. You perhaps may wish to consider the possibility of preparing a little article that will clarify our position and perhaps correct, insofar as your readers are concerned, the impression they may have gained from your editorial that, first, these standards are yet to be substantiated by added field experience and, second, that they rigidly fix the setting heights for any size stoker in accordance with the chart and curves established.

We felt that we were more than ordinarily cautious in the preparation of the release explaining these standards and, as I stated above, we sincerely appreciate your editorial comment and the compliments you paid our Association in our standardization work. This particular recommended standard, as well as other standard codes which we have established, are all subject to revision as necessity or development of the art dictates. We have constantly insisted that it is impossible for any smoke inspector or engineering society or stoker manufacturer or installer to establish a rigid set of curves that would govern setting heights. This was all taken into consideration in our studies on this subject.

(Signed) *Marc G. Bluth,*
Secretary.

Stoker Manufacturers' Association,
307 North Michigan Avenue,
Chicago, Ill.



Addition to the Hanes Hosiery Mills Company plant.

Hanes Hosiery Mills Company

increases its output to over five million dozen pairs of stockings per year by the addition of a new building in Winston-Salem, N. C. Evaporative cooling used on top floor to overcome extremely high sun effect and high occupancy load.

THIS INSTALLATION

Owner—Hanes Hosiery Mills Co.

Location—Winston-Salem, N. C.

Architects and Engineers—Albert Kahn Associated Architects and Engineers, Inc., Detroit.

THE Winston-Salem, N. C., plant of the Hanes Hosiery Mills Company is the largest mill in the country making women's circular knit hosiery. Since the erection of the new addition to the plant described in this article, the capacity of the Hanes Mills is 62,400,000 pairs of stockings a year, as compared with an output of 36,000,000 pairs annually before the new addition was built. Many different styles of hosiery are made by this company of both silk and rayon, including full and knee-length, and ranging from crepe chiffon to service weight numbers.

The new addition, which increases the floor space from 215,000 sq. ft. to 405,000 sq. ft., was required because of the growth of business. Manufacturing operations, beginning at pairing and ending at shipping, have been transferred from the old building into the addition.

PROFITS IN AIR

Activities in the addition include inspection, labeling, packaging and storage. It is a three-story, 250 x 250 ft. building connected to the original mill at the third floor by a covered passage-way so that the flow of goods is from the top to the bottom.

The first floor is at present not in operating use and is being utilized as a warehouse. The second floor, which is level with the street on the east side, is occupied by storage, packing, and shipping departments primarily and also the company's printing department. Since the first and second floors require a minimum of light they are designed with sufficient window area to enable work to be executed under ordinary daylight conditions. The third floor is used for sorting, inspecting, and labeling operations and daylight here is obtained through top monitor lighting.

The first two stories are of reinforced concrete with mushroom type circular columns while the third story is of structural steel H columns and bent beam trusses. Walls are of red face brick with interior of unglazed tile and steel sash.

Heating

Steam heating is used through the building, steam being supplied from the existing boiler house which is connected to the new building through a 3-in. high pressure main. This steam is reduced in two stages to 5 lb. per sq. in., carried over the bridge from the old building and fed to the unit heaters, radiators, and air conditioning units throughout the addition.

Condensate return lines from the heating apparatus are connected to a condensate pump and receiver in a pit at the south side of the building from whence it is pumped to the third floor ceiling and across the bridge connecting the addition to the old mill, from which the condensate flows back to the feed water heaters in the boiler house.

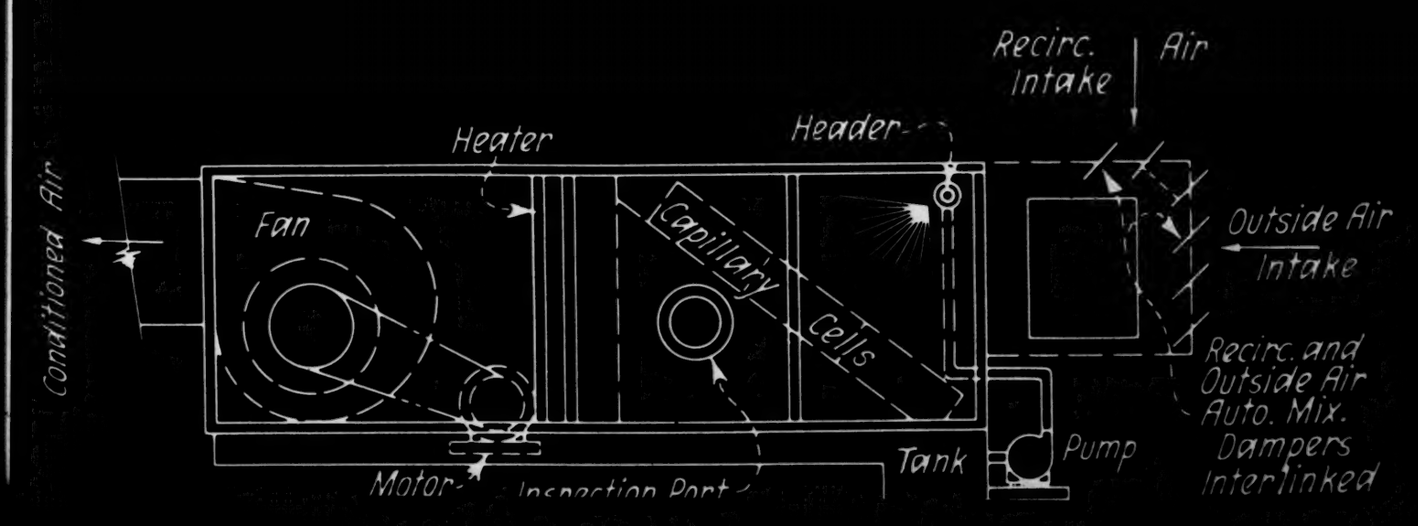
On the first and second floors unit heaters are located to distribute the heat evenly through the manufacturing areas.

On the first floor the unit heaters are of the conventional propeller type. On the second floor projection type heaters with rosette type outlets were installed in order to eliminate air drafts as much as possible. On both of these floors cast-iron radiators are used in the locker rooms, washrooms, and certain other locations around the walls.

Air Conditioning

On the top or third floor about 500 girls are employed. The sun effect here is extremely high, and it was estimated that without air conditioning a maximum inside temperature of 110F would have resulted in the summer. Consequently, it was considered desirable to arrange for more adequate air conditioning on this floor. Four 20,000 c.f.m. Air and Refrigeration Company's capillary units were installed with the idea of maintaining, by evaporative cooling, an indoor condition at least no higher than that outdoors. A further degree of cooling can be obtained merely by supplying the capillary units with low temperature water obtained from a well or cooled by means of refrigeration.

Fig. E. Sketch showing one of the Air & Refrigeration Corp. evaporative cooling type air conditioning units.



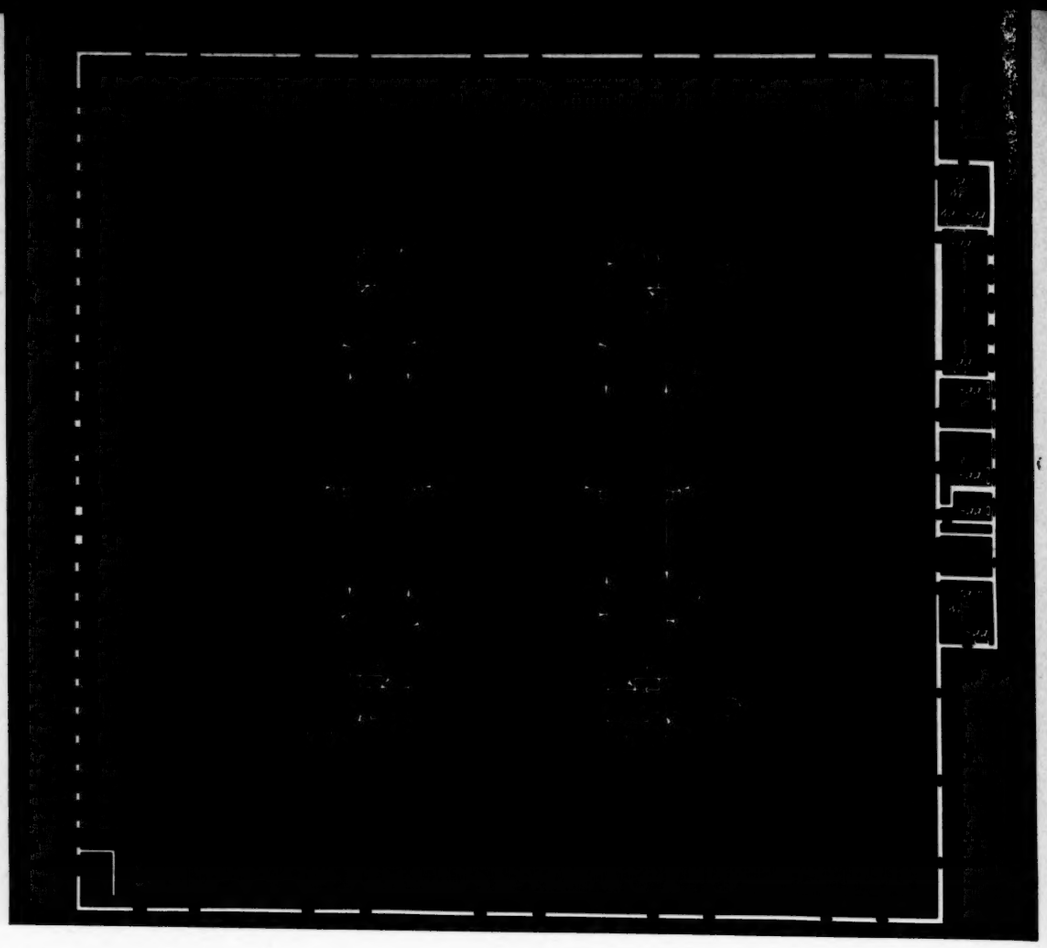


Fig. D. Sketch of the third floor of the Hanes Hosiery Mills addition showing location of the air conditioning units.

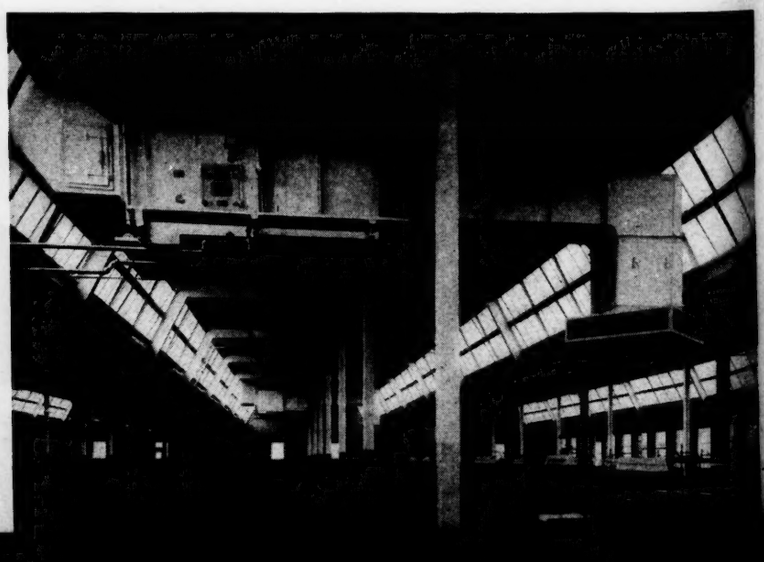
In addition to the cooling functions a constant relative humidity is maintained in the winter by the capillary units. This is desirable in connection with the processes being carried out on the third floor since a relative humidity of approximately 55% facilitates inspection of the stockings.

The four air conditioning units installed are equispaced as shown in Fig. E. Since there are only four units, each unit is conditioning 47,500 sq. ft. of space without objectionable drafts and the distribution of air is reported to be surprisingly uniform due to the method employed to introduce the conditioned air into the space. The method used is illustrated in Fig. C which shows the outlets on the end of the duct. These outlets consist of a rectangular closed box with grilles with directional vanes making up the sides. These vanes are adjustable in both horizontal and vertical planes. The outlets are placed on the duct at an angle of 45° so that the cone of air from the outlet will adequately supply the corners of the building. The air is changed approximately every 10 minutes, excess air being removed through balanced louvers in the wall.

Fig. C shows how valuable floor space was saved by mounting the units in the monitor space so that the air conditioning units are entirely above the working space. The only part of the conditioning system which approaches the working space is the duct outlets which are 12 in. below the steel work. Consequently not only was space saved but there was practically no obstruction of light due to the air conditioning system.

Fig. B. Projection type unit heaters are employed on the second floor of the Hanes plant.

Fig. C. One of the air conditioning units in the foreground. Another unit can be seen in the background.



PROFITS IN AIR

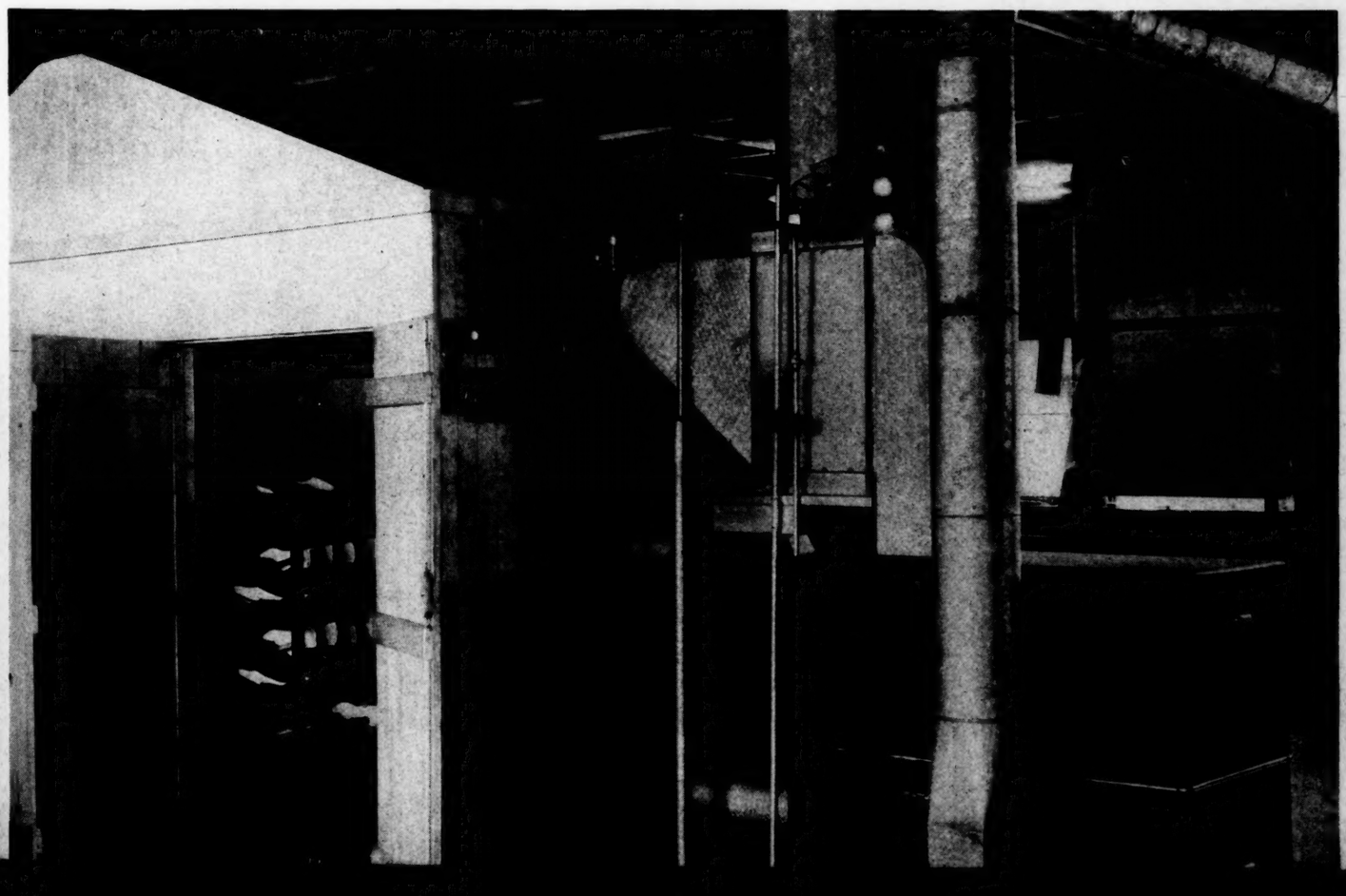
W. B. COON SHOE COMPANY,

Rochester shoe manufacturer, dehumidifies shoe drying shed, cuts drying time 90%, speeds up entire production line and eliminates sole discoloration.

IN shoe manufacturing, after the sole has been sewed to the upper, the shank is trued and the sole contour shaped. This operation is followed by the recementing of the laid-back leather edge over the sole stitching. For the subsequent process the shoe (on its last) is kept moist with wet burlap in order to make the leather workable and the conformation easy. After this operation is completed the burlap is removed but the leather, being hygroscopic, has absorbed a considerable amount of moisture, so that at this stage the shoe is placed in a drying shed and dried out before the shoe is subjected to the leveling, sanding, buffing, imprinting, and final finishing.

In the plant of the W. B. Coon Shoe Company, which manufactures 25,000 pairs of women's shoes per month, the drying was accomplished by letting the shoe stand as long as 24 hours in the open atmosphere of the factory. In such cases it can be seen that the number of shoes in process was kept at an extremely high level due to the large number of shoes accumulated in the drying room. In order to overcome the weather variable in drying, reduce the number of wooden lasts simi-

Fig. 1. The dehumidifyer in the W. B. Coon Shoe Company plant which supplies 600 c.f.m. at 30% relative humidity to the drying room at the left.



larly tied up and step up the drying speeds, this company three years ago installed a gas-fired dehumidifier in a wooden drying shed capable of handling 240 pairs of shoes at once. The result has been to cut the drying time 90%, obtain uniform drying and partially reduce the inventory of those in process.

The heart of the installation is a gas fired silica gel unit, which dries air by passing it through beds of silica gel and keeps these beds fresh by reactivating them when they have absorbed their fill of moisture. The unit consumes 130 cu. ft. of manufactured gas per hour at capacity, and has a connected electric load for operating fans and blowers of 0.6 kw.hr. Attached to the dry air outlet of the unit is a standard after-cooler, a fin-type heat interchanger chilling the air to the desired point by cold water. The assembly operates without attention except for oiling.

Air at 30% relative humidity and 90F is delivered from the dehumidifier-cooler unit high into the end of the drying room at a rate of 600 c.f.m. From the same end of the drying room, 500 c.f.m. of air is exhausted near the floor line and returned to the silica gel unit where the moisture it has picked up from the drying shoes is extracted before recirculation.

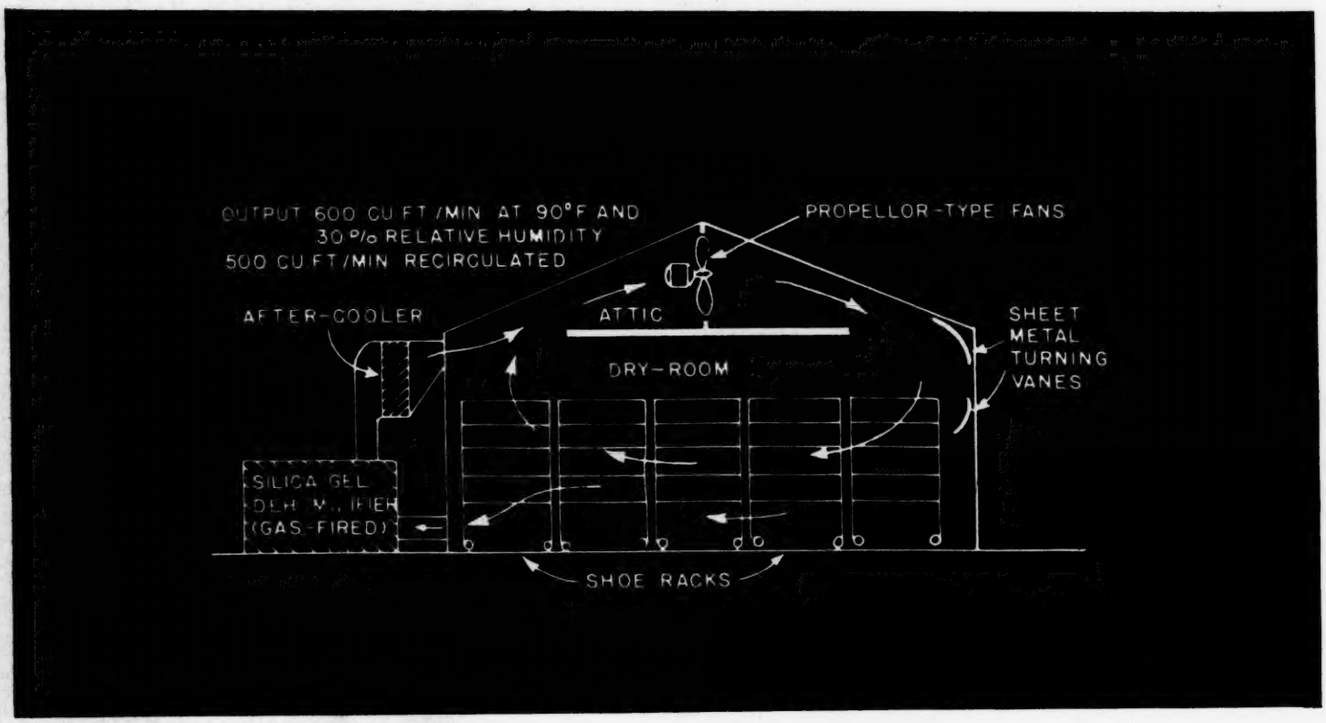
The room itself is approximately 10 ft. by 15 ft. with a cubical content of 896 cu. ft., and is assembled (with a gabled roof) from 1¼ in. tongue-and-grooved pine boards, painted inside and out with aluminum paint for maximum moisture-tightness. After considerable experimental work by Charles King, plant engineer, it was found necessary to install a horizontal baffle of wood, 6 ft. 3 in. from the floor and extending to within 3 ft. of either end of the dry-room to provide the equivalent of a little attic in which is located two 30-inch propeller-type fans recirculating the air through the load of racked shoes.

Independent control of both humidity and temperature conditions around the shoes is provided through a humidistat located inside the dry-room, four feet from the floor and directly connected to the silica-gel unit, as well as by a thermostat which actuates an automatic motor-driven water-control valve in the after-cooler circulating system.

The silica gel unit and after-cooler are operated in muggy summer weather, but when normal winter dryness prevails the unit only operates when the moisture from the shoes raises the humidity in the recirculated air above the humidistat setting of 30%. Therefore, the air conditioning equipment is only operated on an average of 700 hours per season.

With the present arrangement, shoes are now let stand only 20 or 30 minutes (to prevent superficial *flash* drying of the surface and the locking-up of internal moisture when the shoes first go into the dry-room), and then are put into the dry-room just described for no more than one hour—it having been ascertained that 45 minutes at 30% relative humidity and 90F is required to remove the bulk of the leather moisture and 15 minutes more is required to temper the dried sole. As a result, when shoes are now removed from the dry-room they are in perfect condition for leveling, and in better condition for the subsequent sanding and buffing operations which cannot be properly done without revealing discolorations or raising undesirable nap unless the drying is extremely uniform and complete. Trouble has been minimized in leveling, sanding and buffing operations since the gas-fired dehumidifier was installed.

Fig. 2. Sketch of the drying room showing its arrangement relative to the dehumidifier. The propeller fans circulate the air.



LOW VELOCITY EXHAUST SYSTEMS

make possible a considerable saving in power and maintenance costs. The basic principles of such a system and a typical installation in porcelain insulator plant are here described.

By THEODORE HATCH

Formerly with Division of Industrial Hygiene, New York State Dept. of Labor

IN the common dust exhaust system a high air velocity is required in the exhaust pipes to convey the collected dust from the several exhaust hoods to the dust collector which may be a considerable distance from the hoods. The carrying velocities most commonly specified for industrial dusts range from 3000 to 4000 f.p.m. In order to develop a velocity within this range in the pipes the fan must operate against a considerable resistance with consequent high power consumption and costly operation. The relatively great pipe resistance also makes it difficult to balance the air flow through the system and this frequently results in insufficient flow into hoods remote from the fan to give proper dust control at these points. The reduced air flow may also decrease the pipe velocity dangerously and lead to dust settlement in the system which will eventually plug the pipes. Another objection to high pipe velocities is that elbows and even straight sections of pipe wear out rapidly when abrasive dusts are being conveyed. A high entrance velocity into branch pipe connections to hoods will, in certain cases, result in serious loss of the commercial product itself.

Because of these serious objections, considerable effort may profitably be directed toward the development of exhaust systems that do not make use of pneumatic conveying of the collected dust and thus will permit the selection of any convenient and economical pipe velocity with consequent saving in the cost of power and maintenance.

The method that has been employed successfully in one manufacturing plant† to overcome the foregoing objections is described in the present paper. It is of particular interest because the principles used have wide application and do not involve great expense. This method of designing exhaust systems can be applied in small and large plants with equal profit.

†Lapp Insulator Company, LeRoy, N. Y.

In the two exhaust systems to be described, the dust-producing operations are grouped around central dust-collecting chambers with the exhaust hoods connected directly into their side walls. The bottom of the hoods slope downward in each case (with two exceptions) at an angle of 45° or more so that the dust falls by gravity into the collecting chamber and pneumatic conveying is not required. Hence it is not necessary to employ branch pipes of small size to create high pipe velocities. Instead, the hoods are maintained at maximum cross-sectional area, as shown in Figs. 1 and 3, and the air velocities at the hood faces become the maximum in the systems.

System No. 1 provides ventilation for a series of shaping and cutting processes, a sanding belt and a lathe which are employed for shaping and dressing porcelain insulators. For the shaping processes, a minimum face velocity of 1200 f.p.m. was used as the basis of design. This is approximately equivalent to that developed in standard grinding wheel hoods operating with 2.0 in. static suction in the branch pipe. In the present case, however, the static suction required in the central chamber to create a velocity in excess of 1200 f.p.m. was only 0.27 in. or one-seventh of the usual value.

System No. 2 is used to ventilate a group of twenty-two lathes employed for the shaping of porcelain insulators before firing. For efficient dust control a minimum air velocity in the zone of dust generation of 150 f.p.m. was found to be necessary and with the hoods shown, this required a velocity at the hood opening of 200-300 f.p.m. which is created by a static suction within the chamber of approximately 0.01 in. as shown in Fig. 2. Again, to emphasize the advantage gained, this may be compared with the usual static suction requirement of 2.0 in. at the hood for a high-velocity exhaust system.

Dust control was effective with both systems since

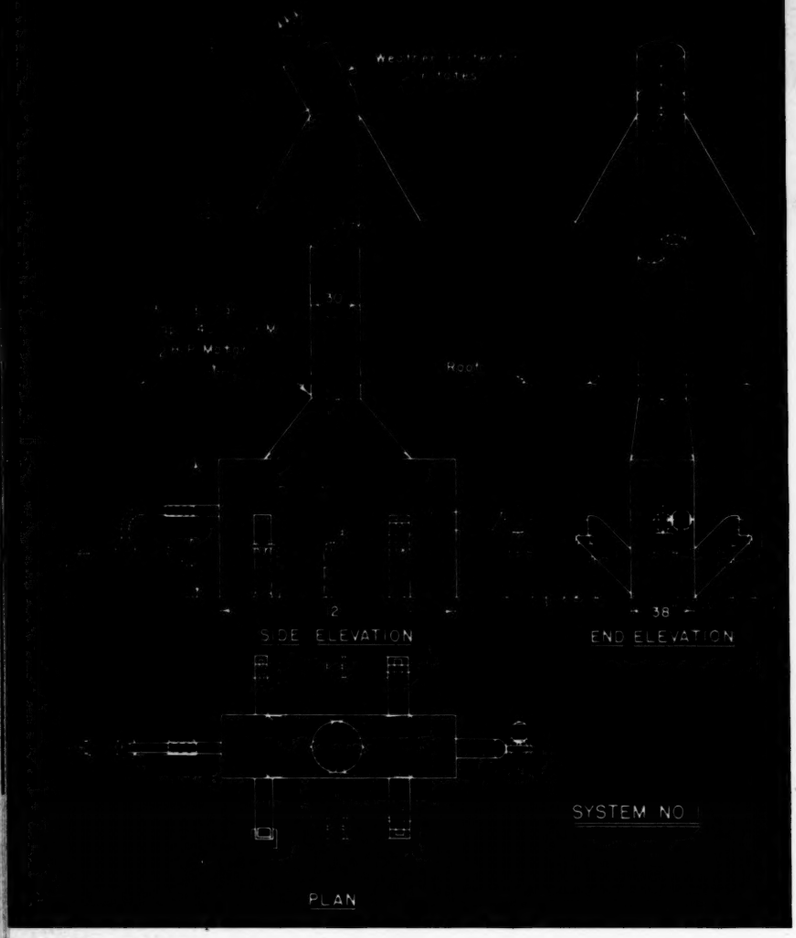


Fig. 1. Low velocity exhaust system for porcelain cutting and shaping operations. For the shaping processes a minimum face velocity of 1200 f.p.m. was used as a basis of design. In this system the air velocity at the hood face is the maximum in the system.

the dust concentrations to which the operators were exposed were found to be below 10 million particles per cubic foot. It is to be noted, however, that the inward velocity of 150 f.p.m. at the lathes connected to System No. 2 was not sufficiently strong to draw in coarse cuttings. The shaping tool was therefore provided with a rubber baffle which is manipulated by the operator so as to guide the cuttings directly into the hood. Some material escapes and falls to the floor and this is removed at frequent intervals by a vacuum cleaner. Spillage was said to be no greater with the present installation than was the case with the previous high-velocity system of conventional design.

Since the negative pressure is essentially uniform within the central chambers the air flow through each hood is wholly determined by the shape and size of the hood alone and no difficulty is experienced in securing a balanced system, thus eliminating the trouble so often encountered in the conventional exhaust system with its unequal static suction values at the various hoods.

Dust Collection

The central chambers around which the machines are grouped serve as settling chambers for the removal of the collected dust from the air before it is discharged from the plant. They are of such dimensions that the vertical velocity does not exceed 200 f.p.m. thus insuring a high percentage of removal of the collected material. That portion of the fine particles that escape settlement are finally discharged from stacks about twenty feet above the roof. The stacks are provided with special terminal fittings that prevent the entrance of rain and snow into the chambers but at the same time allow the dusty air to escape in a vertical direction, thus providing for maximum dispersion into the general atmosphere. This is in contrast to the action of the usual canopy-type of weather cap which interrupts the vertical flow from the stack and causes the air to escape horizontally and even in a downward direction. The effectiveness of the dust collector and weather protector are attested to by the complete absence of accumulated dust on the roof after several years of operation.

Since the plant under consideration was well isolated from other industrial establishments and residences, a certain amount of outside pollution was permissible. Such would not be the case in a built-up area and more efficient dust collection would be required. It is a simple matter to incorporate filters into these systems, however, without impairing their basic design.

The chamber in No. 1 System has a flat bottom upon which the dust settles and the collected material is removed by hand shovel through a door in the side wall. The fan is operated during this period and thus provides ventilation to prevent the escape of dust into the plant atmosphere. The workman wears a respirator. Manual cleaning is acceptable in this case because the dust load is relatively light. System No. 2, however, collects several tons of cuttings and dust in a single day which must be removed continuously and for this purpose the chamber is equipped with a mechanical conveyor beneath a hopper which runs the full length of the chamber. The collected material is carried to one end of the chamber where it is elevated into a storage bin from which it is discharged into a closed lorry and returned to the mixer to be used once again in the manufacturing process. Dust disposal, in this case, is

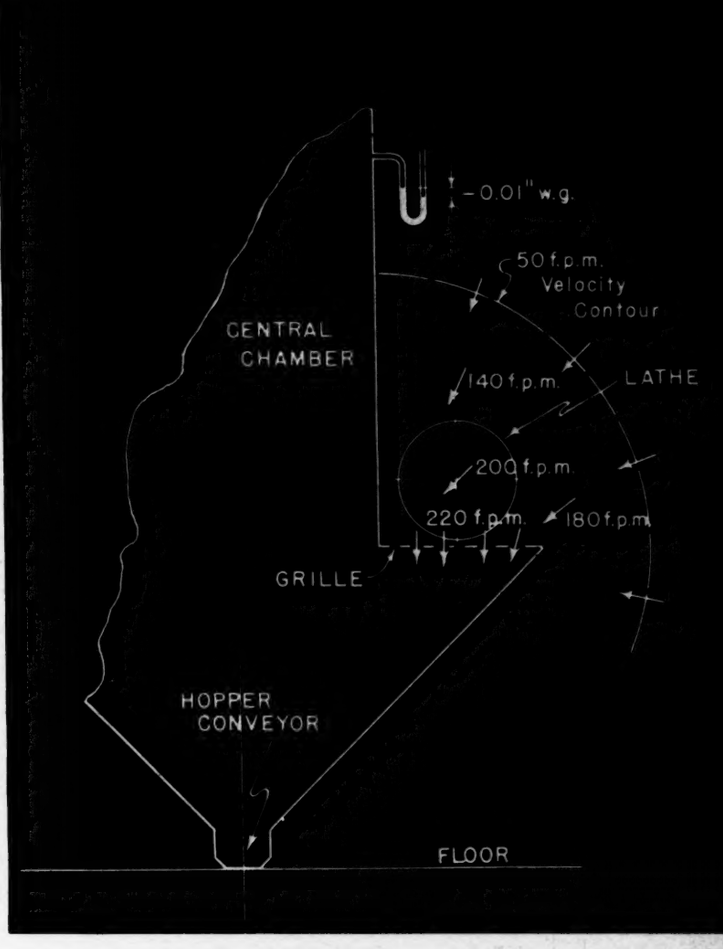


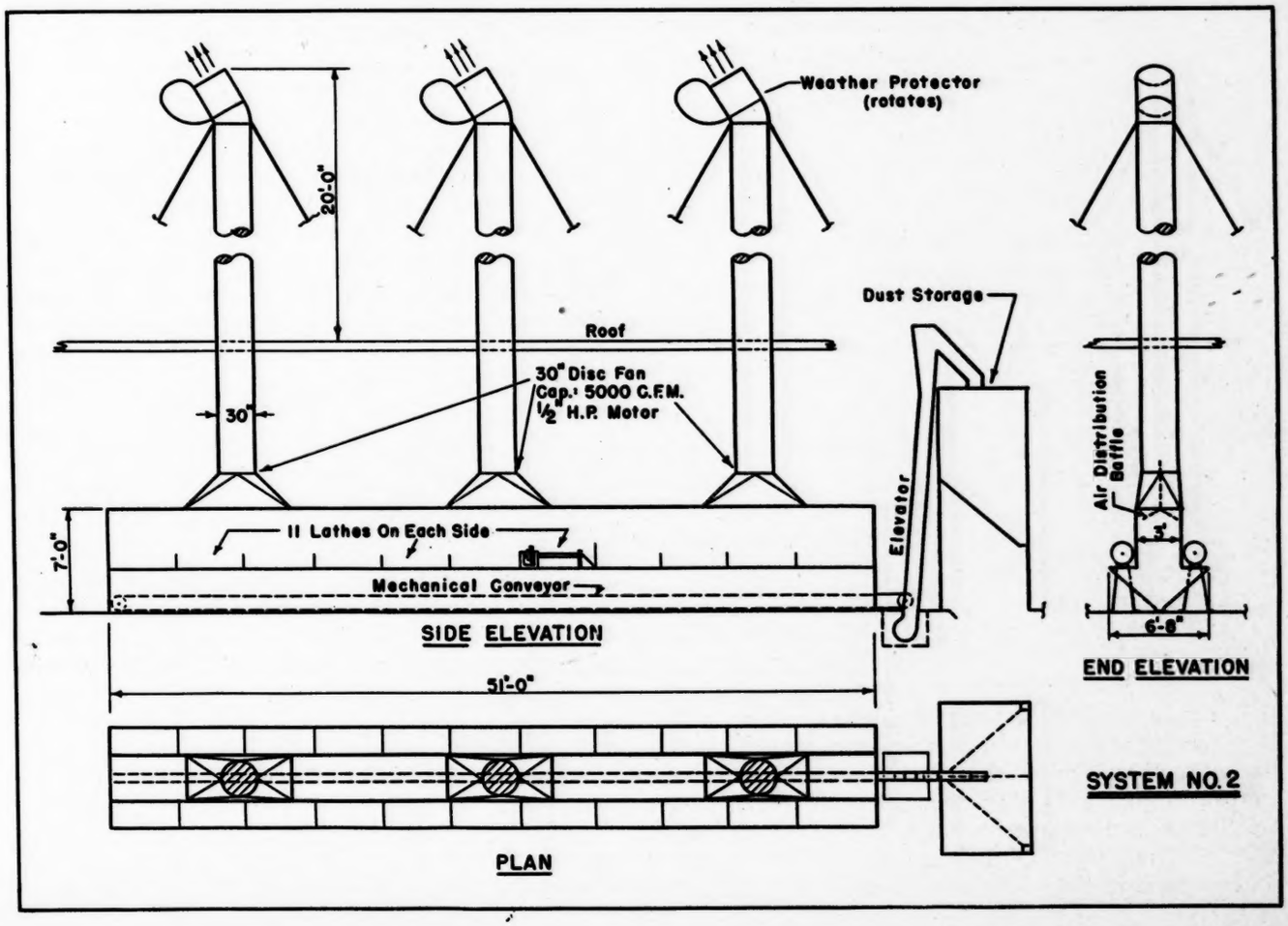
Fig. 2. Air velocities in zone of work in the lathe ventilation system.

accomplished without contaminating the plant or exposing the workmen.

Power Consumption

The total rate of air flow in System No. 1 is approximately 4000 c.f.m. and the overall resistance is less than ⅜ in. Only a ½ hp. motor is required to operate the 30 in. disc fan used in this system. System No. 2 exhausts about 15,000 c.f.m. against a resistance of less than ¼ in. Three 30 in. disc fans are employed, each

Fig. 3. Low velocity exhaust system for ventilating a group of 22 lathes employed for shaping of porcelain insulators before firing. For efficient dust control a minimum velocity in the zone of duct generation of 150 f.p.m. was found necessary and with hoods shown this required a velocity at the hood opening of 200 to 300 f.p.m.



driven by a ½ hp. motor, making a total of 1½ hp. In addition, the mechanical conveyor is provided with a 1½ hp. motor which must be charged to the dust collecting system.

In striking contrast to the low power consumption in these cases a conventional exhaust system in place of System No. 1 would require a 7½ hp. motor while for the lathes 22 hp. actually was used with the earlier ventilating system which exhausted only one-half as much air. Thus, there has been a reduction in power consumption to about one-fifteenth of that ordinarily required. Of equal economic importance has been the saving in maintenance cost. No significant repairs have been required on System No. 2 during four years of operation in spite of the heavy dust load being handled. This is in contrast to the frequent replacement of elbows and piping in the previous high-velocity system.

The marked saving in power effected in this plant will not be realized in every case when these principles of design are employed, especially when a high degree of dust collection is required with its accompanying resistance. The other advantages of lowered maintenance cost, balanced flow, and freedom from the danger of plugged exhaust pipes due to dust settlement will obtain however in every case.

The low velocity exhaust system has other important advantages which may be summarized as follows:

1. Because of the low suction within the chamber, the efficiency of the system is not seriously affected by leakage. Even with the door at the end of the chamber in System No. 2 open, one notes little or no decrease in dust collection. In contrast to this, leaks into worn or damaged pipes in a high velocity system may greatly reduce the air flow through the exhaust hoods and thus destroy their effectiveness.
2. With the dust-generating processes grouped around the central chamber there is a concentration of the ventilation and an orderly movement of air toward the inlets. The chamber acts as a screen which minimizes the dissemination of dust through the room by air currents.
3. The low velocity system can be enlarged to include more machines without having to replace piping. It is necessary only to extend the chamber and install an additional fan or increase the speed of existing fans, if this is permissible. This is a very real advantage over the conventional system with its tapered pipes designed to carry a fixed amount of air only. Once such a system is installed no additions can be properly made without replacing the exhaust main with new and larger pipe. Exhaust systems have failed to function effectively more often because of overloading than for any other reason.
4. The total air volume removed in the low velocity system can be reduced when all the machines are not in use without fear of dust settlement and plugging of pipes. In a high velocity system, on the other hand, all hoods must remain open in order to maintain an adequate transporting velocity in the exhaust pipes. This is an important advantage when heating costs are high due to the removal of large quantities of air.
5. The system under consideration offers another advantage in those cases where the collected dust is to be returned to the manufacturing process. Because of the low velocities employed there is little danger of reducing the size of the particles by attrition and thus spoiling the material for re-use. This has been demonstrated in the present case with System No. 2 which replaced earlier high-velocity ventilation. The collected material is said to be much more valuable than was that removed from the previous collector.

A serious disadvantage of the central plenum chamber design is that it is not directly applicable to scattered machines. The re-arrangement necessary to bring them within reach of the chamber may be contrary to the production requirements.

Other Applications

The principle of low-velocity ventilation has direct application to many processes involving the handling of finely powdered materials which are readily picked up and removed by the ordinary ventilating system and thus lost from the manufacturing process. In such cases the central chamber may be located directly above the hooded processes with vertical pipes of large diameters connecting to the hoods. Low pipe velocities are employed to minimize the removal of the product and the central chamber serves further to trap the material which is then returned to a conveyor belt, elevator or other suitable part of the manufacturing process.

This method of ventilation is particularly useful when the processes are well enclosed and the rates of air flow from the various operations cannot be determined in advance of plant operation since they depend upon the tightness of the enclosures. Under this condition, the conventional exhaust system may fail to operate properly owing to errors in estimating the air flow from the various enclosed processes. With too little air flow from tightly enclosed operations inadequate transporting pipe velocities and dust settlement will result. Such a situation can be corrected only by providing air intake openings in the ventilating system itself. This means an unnecessary wasting of power. When pneumatic conveying is not employed, on the other hand, the vertical connecting pipes can be made of any reasonable size without fear of plugging. Such a system will automatically adjust itself to the varying amounts of leakage into the enclosed operations and any settlement that does occur will cause the dust to return to the production line. It should be pointed out however, that the self-cleaning of vertical pipes is not to be depended upon when the dust has cementing or adhesive properties. In such cases, high transporting velocities or mechanical conveying is required.

Many other applications will suggest themselves when the advantages to be gained from low-velocity exhaust ventilation are realized.

Acknowledgment

The writer gratefully acknowledges the permission of the Lapp Insulator Company to publish a description of their ventilating equipment and desires to give full credit to them for the developments described in this paper.

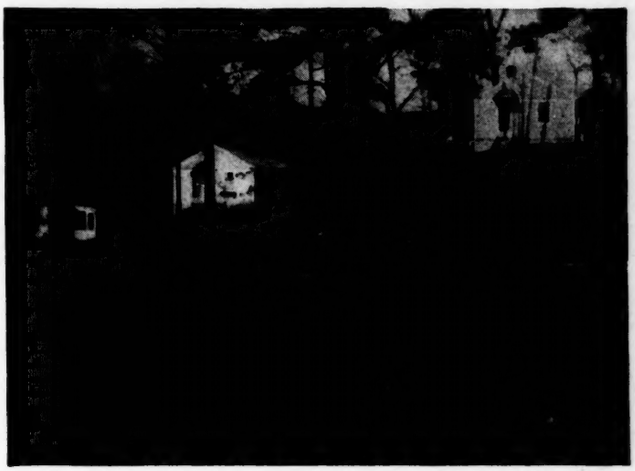
Savings from Insulation in Low-Cost Housing

By W. H. PURNELL†

Last month Robert K. Thulman described on these pages the effect of insulation on comfort. In the accompanying article Mr. Purnell discusses the economic aspects of insulation and reports on the results of tests comparing computed and actual heat losses of insulated and uninsulated houses. This paper is abstracted from the Centennial Meeting of the National Mineral Wool Association recently held at Chicago.

THE Tennessee Valley Authority has for the past several years made a study of small house planning as a desirable adjunct to the building of something over a thousand houses for employees at the various dam-building projects. One of the factors of any such study—and an important factor—is heating.

Many of the first houses built by the Authority, those at Norris, Tennessee, were heated electrically. Since the elements in the heaters made for practically 100% efficiency in conversion of electrical energy to heat units, conductance of tests of thermal characteristics of the structure of these houses was a simple operation. Unoccupied houses were available for testing and the houses could be kept closed and heated continuously at a thermostatically-controlled temperature. Records of inside and outside temperatures over periods of time gave a measure of the degree-hours of heating. Metering of the electrical energy input for heating over the same periods of time gave a record of the heat units required. The heat loss of the entire house was then successfully reduced to B.t.u. required for each degree-hour. Minor variation from day to day in the B.t.u. per degree-hour unit only served as a gauge of the effect of outside weather on heat losses.



TVA low cost housing. The tests referred to were made on similar structures

Tests of the heating in Norris houses were made during three successive winter seasons, and the possible rigid control of test procedures produced considerable data on the direct heat transmission through certain house structures and the thermal environment within habitable spaces. Although these data were of basic importance, they did not entirely fulfill the need for reference material. Pertinent questions on the thermal characteristics of housing remained unanswered. The houses tested were insulated, yet from tests in any one house it was impossible to determine the value of the insulation because the heat loss was a homogeneous quantity for the entire assembly of the elements of the house. True, the heat loss could be computed and the percentage of the total loss for any particular element such as the floor estimated. But this was done and the results were of very doubtful accuracy. The total computed loss of the house did approach coincidence with the loss found by test, but when closing the vent openings to the ventilated below-the-floor space showed a reduction of over 20% in the total heat loss, this analysis was abandoned. The loss through the floors had been reduced by over 20%, yet the computations had not allotted even as much as 20% of the total heat loss to the floors.

However, since the total heat losses, computed and tested, were near agreement, the heat loss of a comparable house of average frame construction and with no insulation was computed and compared with the heat loss of a typical Norris house. From this comparison it was, lacking conclusive data, assumed that the insulation in the Norris house had reduced the heat over what it would have been if not insulated by approximately 40%. Incidentally, the typical Norris house has four inches of mineral wool in the walls and two inches of mineral wool over the ceiling.

With regard to the thermal environment, in the typical Norris house, the temperature differential

†Architect, Department of Regional Planning Studies, Tennessee Valley Authority.

from floor to a 7 ft. 8 inch ceiling was about 10F. This is in contrast to differentials of 20F, 30F, and even higher, found to exist in uninsulated houses in this country. It was a recognized fact that the gravity type electric heaters used locally, one in each room, played a part in maintaining a low temperature differential within the houses tested. It was realized that the mild Tennessee climate was an asset in the maintenance of a desirable thermal environment. Also, it was thought that insulation had a very definite influence on comfort —but influence to what extent was a matter of speculation.

The Study

During the fall of 1938 the Authority decided to make a study which would give conclusive evidence of the value of insulated housing. All houses previously built by the Authority were studied with regard to choice of houses to be subjected to test. The Type D house at the Hiwassee Dam construction community was selected for three reasons. First, this type house was most suitable for the installation of insulation at such a time as normal construction had been completed. The exterior wall surface consisted of weatherproof wallboard in large sheets that could be easily removed. Second, although these houses were not of the normal frame house construction, the computed heat loss of these houses was within 4½% of what the heat loss would have been had the houses been of frame. And last, insulation installed and tested in one of these houses would give an indication of the relation of insulation to low-cost housing, the Type D Hiwassee House costing, net labor and materials, less than a thousand dollars. This type house was constructed without insulation at this particular community because the period of amortization was only three years.

All houses of the type selected were occupied and the actual choice of houses for use was made from a list of houses whose occupants had offered to cooperate. These houses are 20 feet by 26 feet, basementless, and contain four rooms and bath. Two adjacent houses with identical orientation and equally severe exposure were used, one with its normal construction intact except for the addition of weatherstripping upon the two doors, the other insulated as well as having the doors insulated. The insulation consisted of ¾ in. insulation board under the floor joists and 4 in. mineral wool batts in the walls and over the ceiling. The type of insulation used was a matter of arbitrary choice. The moisture barrier on the warm side of the wall that would have been used in a locale of more severe climate was eliminated.

Comparison of computations of heat losses of the two houses, insulated and uninsulated, showed a reduction in heat loss effected by the insulation of approximately 60%. This figure would of course apply only when the two houses were continuously heated and does not take into consideration any heating cycle whereby the house is either not heated or only partially heated at night. As it turned out, the daily living cycles of the two families in the houses were quite divergent and it was necessary to adopt a schedule whereby the heat would remain on during the night in all rooms except bedrooms. By adopting such a schedule the two houses could be continuously heated except for bedrooms. Heat was turned off in the bedrooms of both houses at 9:30 o'clock each night and on again at about 7:00 o'clock each morning. Bedroom doors were closed during the night, bedroom windows opened. The only other requirements of the families was that they leave outside doors and windows closed, avoiding unnecessary holding open of doors when entering or leaving the house.

For testing purposes, electric heating was temporarily installed, with a 2-kw. portable heater in each of the four rooms. The heating load could then be accurately measured through a meter separate from the one used to meter the load for lighting, cooking, and appliances. This meter was of the recording demand type, recording the demand in kilowatts at fifteen-minute intervals upon a 7-day chart.

After the testing set-up was completed, the heaters and instruments were used for several days prior to the actual tests in order that the families and the test operators might become thoroughly familiar with the testing routine. Readings taken in the houses previous to the installation of the electric heaters showed that these families, using coal-circulating heaters, had become accustomed to room temperatures as high as 90F. This preliminary period was useful in that it allowed the operators to reduce the room temperatures gradually, without harm to the occupants, to normally accepted levels. Tests were then begun and continued for fifteen consecutive days.

Comparison of Results

A comparison of the heat losses of the two houses at the end of the testing period showed that the insulation applied to the normally uninsulated Type D house had reduced the heat loss of that house by 44.75%.

At first thought the 44.75% saving shown by test, when compared with the 60% saving indicated by computation, appears to be somewhat below expectations. However, further analysis of the data points out the full significance of the figure. As said before, the 60% saving indicated by the computations was based on uninterrupted heating. The houses tested were only partially heated during the night. When the bedrooms were closed at night, with heat off and windows open, a fraction of the heat loss was through the partitions separating the heated space and the unheated bedrooms. Also, infiltration of cold air through the half-inch crack under the closed bedroom doors accounted for part of the heat loss during the night. Consequently, it was found that the ratio of the heat losses of the two houses was slightly different during the night from what it had been during the day when the entire house was heated. Comparing the daytime heat losses of the two houses, 7 a.m. to 9:30 p.m., we find the insulation showing a reduction in heat loss of 46.23%. However, the heating during this period includes that required for the morning pickup—that absorbed by the structure. Therefore, this figure is still not to be compared with the 60% computed saving.

Going further than this, the heat losses of both

houses appeared to have at least approached equilibrium by noon. Comparing the heat losses of the houses for the period from noon to 9:30 p.m., making adjustment for the somewhat higher temperature in the uninsulated house during this period, we finally arrive at a reduction in heat loss by insulation during periods of continuous heating of approximately 55%. This 55% saving then is the real figure for comparison with the computed 60% saving originally estimated. I believe anyone who knows the assumptions that must be made and the chances for error when computing house heat losses from coefficient tables will agree with me when I say that estimates within 5% are satisfactory.

You may by now be ready to question the value of such an analysis whereby a saving of 44.75% shown by test is built up to 55%. Obviously, no house under conditions of normal occupancy would be 100% heated night and day, and the 55% saving would therefore not be effected. The value of the analysis is that only by checking the possible computed savings are we able to have any indication of the effect of house occupancy upon heat losses. In this particular case it appears that the actual saving during occupancy might be taken as about 10% less than the saving shown by comparing computed savings for periods of continuous heating.

Economic considerations of thermal insulation depend upon climate, heat loss of uninsulated structure, occupancy, life expectancy of building, cost of producing heat, cost of insulation, and other factors. Since the Type D house, uninsulated, has a heat loss comparable to a typical plaster and wood siding house, it therefore appears that a generalization of the economics of insulating this house might be significant.

Economic Analysis

The insulation installed in the Hiwassee house cost, labor and materials, $200. This expenditure included removal and replacement of all the exterior wallboard panels of the house as well as some required sub-floor excavation. Had this insulation been installed at the time of building the house, it is estimated that insulating and weatherstripping of doors only would have cost not more than $113. In order to examine the economics of spending this amount to insulate this house, we shall assume three different conditions of climate and calculate the yearly dividend in fuel savings paid by this $113 worth of insulation. In addition, but not included in tabulations below, insulation will pay a dividend in the saving in capital cost of heating equipment, equipment of less capacity being required. I shall say more of this a little later. Tests showed that 5755 B.t.u. per degree-day were required to heat the uninsulated house while only 3165 B.t.u. per degree-day were required to heat the insulated house, a saving of 2590 B.t.u. per degree-day. In the locale of Hiwassee, N. C., with an approximate 3800 degree-day heating season, we see that the yearly dividend paid by the insulation is

$11.52 for heating with electricity @ 4/10c per kw. hr., TVA rate for 1000 kw. hr. immediately following first 400 kw. hr. used for cooking, lighting, and water heating.

7.87 for heating with oil @ 8c a gallon.

$3.78 for heating with coal, hand-fired, @ $6 a ton.

Therefore, on this basis, we see that the insulation would pay for itself in 9.8 years with electric heat, 14.4 years with oil heat, and 30 years with coal heat. Assuming that the house is amortized for 20 years it is apparent that, with electric heat, the insulation would not only pay for itself but reduce the heating costs by 23%. Likewise, with oil heat, the insulation would pay for itself and reduce the heating costs by 12.5%. With coal heat it is evident that this amount of insulation at this cost could not be economically justified unless capital costs of equipment were so reduced as to bring payment of insulation costs within the amortization period. We will examine this later.

Assuming a hypothetical northern locale with a 6000 degree-day heating season, we see that the yearly dividend paid by the insulation is

$45.50 for heating with electricity @ 1c per kw. hr.

$13.20 for heating with oil @ 8½c a gallon.

$11.93 for heating with coal, hand-fired, @ $12 a ton.

The insulation would pay for itself in 2.5 years with electric heat, 8.6 years with oil heat, and 9.5 years with coal heat. Heating cost with electricity would be reduced 39%, with oil by 25.5%, and with coal by 23.5%.

It is a rather pleasant realization on the part of any prospective home owner — the realization that in a majority of cases he can build with insulation and reap a premium in fuel savings on his investment. But, fuel savings are not the only inducement for the home builder or home owner to insulate. When the heat loss of any house is reduced by 44.75%, as was the one at Hiwassee, the size and cost of the heating equipment drops accordingly. If the Hiwassee house was built as a permanent residence with some common method of furnace heating, $50 would be a modest estimate of the saving in the capital cost of the heating equipment. This means that $50 of the $113 for insulation has been paid before the house was occupied. Looking back to one of the cases in fuel saving cited, the man whose fuel savings would only pay for insulation in thirty years, can now pay for the insulation within his 20-year amortization period and reap a small premium also. Then we see that reductions in the capital costs of heating equipment may tip the scales at an otherwise undecisive point.

The TVA has during the past two years made a study relative to the design of a warm air, oil-fired, furnace that could be placed in a small chamber on the first floor of a low-cost house, eliminating the need for a basement. This unit has reached a fairly satisfactory stage of development and in the last year has been placed upon the market by a number of manufacturers. The TVA has just finished installing ten of these furnaces in houses at its Kentucky Dam construction village, the unit fitting into a chamber only 3 feet by 3 feet 6 inches. If these houses had not been insulated the capacity of the unit would have had to be increased and the unit probably would have been so large as to have required a basement. Basements, even in low-cost houses, are worth from 200 to 400 dollars and the difference in cost of the three by three feet six inches heating chamber and a basement will pay for the insulation before the house plan leaves the drafting room. The fuel savings are then entirely profit.

Theory vs. Practice—Why Heating Calculations and Requirements Don't Agree

By T. W. REYNOLDS

It is unfortunate that most technical writing is done by designing and research engineers who have a flair or a liking for such writing and very rarely by field engineers who, perhaps due to the nature of their work, usually have little urge to prepare articles or papers. Consequently most magazine articles, textbooks, and handbooks on heating and air conditioning frequently fail to include certain practical and very important information on certain points which may be almost axiomatic to men in the field. The net result is that experienced field engineers often have a somewhat pitying attitude toward the academic and office-trained engineer. The accompanying brief article is an attempt to bridge the gap between what can be conveniently if not accurately termed theory and practice. It is the first of a number of similar articles which will appear from time to time, each of which will be complete in itself.

ACCEPTED methods of figuring radiator requirements to obtain the size of boiler are all very well and their use is quite desirable for prospective installations where the many safety factors included in such figures will serve to compensate for errors and any contingencies that might arise, but an installation already on our hands is expensive to change and therefore should be approached with caution and with the elimination of all the usual factors of safety. The writer has seen many a report rendered which though technically correct in itemized listings, nevertheless proved quite unsatisfactory when its recommendations were carried out, simply because the engineer failed to keep these things in mind and to get at the real root of the trouble.

In enumerating the various safety factors which contribute to oversizing of equipment, one may well begin with computations involving the infiltration of air which has to be raised from outdoor to indoor temperature and which may be figured more theoretically according to the window crack method, but which method in any event should be checked by the arbitrary air change method. If the gross cubage of the house has been in such figures to obtain the air infiltration load on the boiler, then the estimate is high by 25% for we are only concerned in the space heated which is the net cubage of the house and this will usually be found to be 80% of the gross, after deductions for floors and walls are made.

If, on top of this, we have assumed an air change of one-and-a-half for the entire house we are again in error in so far as boiler size is concerned, because the inleaking air is only on the windward side, and this air, after being warmed by radiators on that side, passes out on the leeward side, so that in the most severe cases, such as where the long side of the house is on the windward side, only one-half of the assumed air change should be figured. In other words, while a one-and-a-half air change per room is the correct figure for use in sizing of the radiators, because each radiator must also contain the surface as required to compensate for air infiltration whenever the wind blows on that side of the house, a three-quarter air change of the net cubage of the entire house is all that is required for boiler size, since the wind can only blow on one side of the house at a time and therefore air infiltration cannot take place in all rooms at the same time.

Of course, in line with accepted practice the assumption of one-and-a-half air change was predicated on each room having windows on at least two of its sides; however, if each room has windows on only one side, the air change would normally be assumed as but one for each room whenever individual room computations are made, so that with about half of the rooms on the windward side the proper figure to use for the net cubage of the house would be only a one-half air change.

Even where each room has a window on each of its two sides, it is not always correct to assume a one-and-a-half air change per room. For example, a smaller window on the leeward side of such a room will not contain enough linear feet of crackage to pass an amount of air equal to the air which would otherwise enter through the larger window on the windward side. We know this to be so due to experiences gained in the ventilation of spaces in the summer months by wind where for best results each air inlet must have a correspondingly sized air outlet, or else the ventilating effect of a large inlet will be reduced to that of a much smaller one. Ventilation by natural means has also taught us that the air inlet should face directly into the wind with the air outlet advantageously located on the side of the room directly opposite the inlet. Unfortunately, in the summer months and fortunately in the winter months, the windows in most rooms seldom comply with these requirements.

There are still other factors which make the extent of infiltration through window cracks not all that it is cracked up to be; thus more recent studies indicate that actual infiltration is only 80% of that which occurs in laboratory tests with wind tunnels, due to the fact that there is at least as much resistance to air leaving on a leeward side as there is to its entrance on the windward side. This resistance causes a pressure to be built up within the house to oppose the pressure of the entering air and therefore reduces the amount of air infiltration to the extent stated.

Speed and Accuracy in Figuring Heat Loss

By RALPH A. KRAUSS†

The author here presents tables and graphs which greatly shorten the time required to make heat loss calculations but at the same time maintain the desired accuracy. His tables concerning infiltration and transmission losses through doors and windows of stock size are particularly ingenious.

TIME was when a heating engineer could make a rough estimate of heat losses, multiply by a large factor of safety, and feel satisfied. Today's competitive conditions put a premium on accuracy, which, however is obtained by somewhat complicated and tedious calculations.

In an attempt to reduce the computations required by the precise method, the author has prepared a number of charts and tables, which are here published for the first time. The use of these devices eliminates all long division and much of the multiplication, permitting quick computation without the aid of a slide rule. These aids should encourage more common use of the accurate method of heat loss calculation.

Windows and Doors

Tables I, II and III give heat losses through standard-size windows and doors, eliminating the calculation of areas and crack lengths. For the sake of simplicity, they are based upon 70F temperature difference and 15 mile wind, but a simple chart permits adjustment to other temperature differences, and multiplying factors take care of other wind velocities.

The first two columns give the width and height of the window or door opening in inches. The opening refers to the outside dimensions of the window or door.

The next column gives the area in square feet. This figure is not used in calculating window losses, but is subtracted from the gross wall area to obtain the actual or net area of the wall structure.

The transmission loss is given in B.t.u. per hour for single and double glass, the latter referring to two separate thicknesses of glass with an air space between. The presence of storm sash fulfills this condition, but "double-strength" glass does not. For single glass, the transmission coefficient is 1.13 B.t.u. per sq. ft. per hr. per degree F., and for double glass, 0.45 B.t.u.

Doors consisting largely of glass or thin wood panels are assumed to have the same transmission loss as windows of the same size. For solid wood doors, multiplying factors are given.

Infiltration loss depends upon a number of factors, including the construction of the window and its fit. The tables are calculated on the basis of standard data.

Infiltration losses through types of windows not given in the tables may be calculated by multiplying the loss shown under "Weatherstripped, Poor," by the factors given in Table V.

Metal windows sometimes consist of part stationary and part movable sections. In this case, count the entire window for transmission loss and the movable part

TABLE I.—HEAT LOSSES, DOUBLE-HUNG WINDOWS

Window Size (Inches)		Area (Sq. Ft.)	Transmission Loss at 70F Diff. (B.t.u. per Hr.)		Infiltration Loss at 70F Diff. 15 Mile Wind (B.t.u. per Hr.)			
					Weather-stripped		Non-Weather-stripped	
Width	Height		Single Glass	Double Glass	Aver.	Poor	Aver.	Poor
16	46	5.10	405	162	350	500	580	1620
20	36	5.00	400	160	330	477	545	1545
20	42	5.84	460	184	360	520	595	1685
20	46	6.38	505	202	380	540	625	1740
20	54	7.50	592	236	420	608	694	1970
20	58	8.06	636	255	440	635	725	2060
22	30	4.58	362	145	310	448	512	1450
22	46	7.03	556	221	390	565	650	1810
22	54	8.25	645	260	430	625	720	2020
22	58	8.85	700	280	455	658	750	2130
24	36	6.00	475	189	360	520	600	1680
24	42	7.00	555	222	390	560	645	1810
24	46	7.67	608	242	410	590	680	1920
24	50	8.33	660	262	430	620	715	2000
24	58	9.69	768	315	470	680	780	2180
24	62	10.33	820	326	490	700	810	2290
26	54	9.80	787	308	465	670	775	2170
26	58	10.45	829	329	480	700	800	2260
26	62	11.20	887	352	500	730	840	2350
28	30	5.85	462	185	360	520	595	1485
28	36	7.00	555	222	390	560	645	1810
28	38	7.40	587	233	400	575	660	1860
28	42	8.16	647	258	420	605	700	1950
28	46	8.95	710	283	435	630	725	2040
28	50	9.75	775	307	460	660	760	2160
28	54	10.50	833	331	480	690	800	2250
28	56	10.90	862	344	490	710	810	2300
28	58	11.25	894	354	500	720	830	2340
28	62	12.05	955	380	520	745	860	2420
28	66	12.80	1013	403	540	775	900	2520
30	42	8.75	695	276	430	625	720	2020
30	54	11.25	894	354	490	710	800	2280
30	58	12.05	955	380	510	740	850	2400
30	62	12.90	1020	406	540	780	900	2520
31	30	6.46	510	204	382	554	630	1790
31	36	7.75	612	244	413	596	680	1930
31	42	9.05	715	285	442	640	730	2070
31	46	9.90	782	312	462	670	763	2160
31	50	10.80	850	340	482	700	800	2260
31	54	11.60	915	365	503	725	830	2370
31	58	12.50	990	394	522	755	862	2450
31	62	13.42	1055	423	543	785	895	2540
31	66	14.25	1120	450	563	815	930	2640
32	54	12.00	950	378	510	740	850	2400
32	58	12.90	1020	406	530	770	880	2460
32	62	13.80	1090	435	550	800	920	2580
34	30	7.10	560	224	405	585	670	1895
34	36	8.50	670	268	435	630	720	2040
34	42	9.90	780	312	455	660	750	2130
34	46	10.85	855	342	475	685	785	2230
34	50	11.80	930	372	495	715	820	2320
34	54	12.75	1010	404	520	755	880	2460
34	58	13.70	1080	430	535	775	885	2500
34	62	14.60	1155	460	565	815	940	2640
34	66	15.60	1230	490	575	831	950	2690
36	46	11.50	910	362	500	723	825	2340
36	50	12.50	990	394	520	750	860	2440
36	54	13.50	1065	425	540	780	890	2530
36	56	14.00	1100	440	550	795	908	2580
36	58	14.50	1145	460	560	810	925	2620
36	62	15.50	1220	488	580	840	956	2720
36	66	16.50	1300	520	600	868	990	2810
40	46	12.75	1005	402	530	766	875	2480
40	50	13.90	1095	438	550	795	908	2575
40	54	15.00	1185	472	570	825	940	2670

†Combustion Engineer, Anthracite Industries Laboratory.

TABLE II.—HEAT LOSSES, SINGLE CASEMENT WINDOWS

Size of Opening (Inches)		Area (Sq. Ft.)	Transmission Loss at 70F Diff. (B.T.U. per Hr.)		Infiltration Loss at 70F Diff. 15 Mile Wind (B.T.U. per Hr.)			
					Weather-stripped		Non-Weather-stripped	
Width	Height		Single Glass	Double Glass	Aver.	Poor	Aver.	Poor
20½	21	3.0	237	95	205	296	346	970
20½	45	6.4	506	202	325	468	545	1530
20½	53	7.54	595	238	363	524	610	1710
22½	23	3.75	296	118	224	324	378	1060
24	24	4.0	316	126	248	343	400	1120
24	30	5.0	395	157	268	386	450	1260
24½	25	4.25	335	134	253	365	425	1190
24½	29	4.92	388	155	265	382	445	1250
24½	53	9.0	712	284	381	550	640	1800
25½	23	4.08	322	129	240	346	404	1130
28½	25	4.95	390	156	265	382	445	1250
28½	29	5.74	452	180	284	410	478	1340
28½	31	6.12	473	193	295	425	495	1390
28½	35	6.92	545	218	314	454	528	1480
28½	41	8.10	640	255	343	495	578	1620
28½	47	9.30	735	293	374	540	628	1760
28½	53	10.5	830	331	403	582	678	1900
31½	29	6.35	500	200	300	430	502	1410
31½	33	7.22	570	228	320	460	535	1500
31½	35	7.66	605	242	330	474	552	1550
31½	37	8.10	640	255	340	490	570	1600
31½	41	8.96	709	283	360	518	603	1690
31½	47	10.3	815	325	390	560	652	1830
31½	53	11.6	915	366	420	603	702	1970
34½	29	6.95	548	219	315	453	528	1480
34½	33	7.90	625	249	335	480	560	1570
34½	37	8.85	700	279	355	510	595	1670
34½	41	9.83	788	310	375	540	627	1760
34½	47	11.2	895	352	405	582	678	1900
34½	53	12.6	995	396	435	625	728	2040

TABLE III.—DOORS AND DOUBLE CASEMENTS

Size of Opening (Inches)		Area (Sq. Ft.)	Transmission Loss at 70F Diff. (B.T.U. per Hr.)		Infiltration Loss at 70F Diff. 15 Mile Wind (B.T.U. per Hr.)			
					Weather-stripped		Non-Weather-stripped	
Width	Height		Single Glass	Double Glass	Aver.	Poor	Aver.	Poor
SINGLE DOORS								
24	78	13.0	1025	410	1190	2380	2380	4760
24	80	13.3	1050	420	1215	2430	2430	4860
28	78	15.2	1200	480	1240	2480	2480	4960
28	80	15.6	1230	492	1250	2500	2500	5000
30	78	16.2	1280	512	1260	2520	2520	5040
30	80	16.7	1320	530	1285	2570	2570	5140
32	80	17.8	1405	562	1310	2620	2620	5240
34	82	19.3	1525	610	1355	2710	2710	5420
36	80	20.0	1580	632	1355	2710	2710	5420
36	84	21.0	1660	665	1400	2800	2800	5600
DOUBLE OR FRENCH DOORS AND WINDOWS								
32½	35½	7.8	615	246	430	620	715	2010
36½	41½	10.5	830	332	490	710	820	2300
36½	45¾	11.5	910	365	505	730	840	2360
36½	53¾	13.6	1070	428	585	845	975	2740
40½	35½	10.0	790	316	467	675	780	2190
40½	41½	11.7	925	370	515	740	855	2400
40½	45¾	12.8	1010	404	550	790	910	2550
40½	53¾	15.1	1190	475	605	875	1010	2830
44½	53¾	16.7	1320	530	625	900	1040	2920
48	78	26.0	2060	825	1925	3850	3850	7700
48	80	26.7	2110	844	1960	3920	3920	7840
48	84	28.0	2210	884	2025	4050	4050	8100
60	78	32.5	2570	1030	2065	4130	4130	8260
60	80	33.3	2640	1060	2100	4200	4200	8400
60	84	35.0	2770	1110	2165	4330	4330	8660

The above figures are for doors consisting of glass or thin wood panels. For solid wood doors, multiply the transmission-loss figures given for double glass by the factors in Table IV.

for infiltration loss. Leakage around doors, because of the tendency of doors to warp, is generally greater than for comparable windows. In the tables, the leakage per foot of crack is the same for well-fitting doors as for poorly-fitting windows; weatherstripping halves this, and poor fit doubles it.

Storm sash and storm doors reduce the transmission loss to the same value as double glass, and often reduce the infiltration as well. For well-fitted or weatherstripped windows, little, if any reduction can be counted upon, but poorly-fitted windows will be reduced about 50% in infiltration loss, as will doors, except those which are weatherstripped, in which case no reduction should be assumed.

Infiltration, Walls and Roofs

Use of building paper, wall-paper, paint or plaster reduces the infiltration through finished walls to a negligible figure. However, for unfinished walls, roofs, etc., use Table VI.

Note that the infiltration through the windward side must pass out through the leeward side. Therefore, for roofs count only half of the total area; for sun porches and sheds count only the area facing the prevailing winds.

Wind Velocity Factors

The tables have been based upon a 15 mile wind because this is perhaps the most commonly used figure. The proper velocity for any given locality (average velocity for December, January, and February) may be obtained from the local Weather Bureau. For wind velocities other than 15 m.p.h., use the multiplying factors in Table VII.

Total Infiltration

The figures in the tables are for the windward side of the house, the same outward leakage occurring on the leeward side, unless there are other means of exit, such as open fireplaces. The basic problem is to find the maximum heat necessary to make up the combined transmission and radiation losses; in most well-constructed buildings it is usually safe to consider that the maximum heat loss occurs when the outdoor temperature is lowest, the infiltration loss at that time being determined by the average velocity and prevailing direction of the wind.

Therefore, infiltration may be assumed to occur only on the windward half of the building, although it is safer to compute the total possible leakage of the entire structure, making sure that the infiltration loss is not less than half of this figure.

Walls, Roofs, Ceilings, and Floors

The usual method of computing the heat transmission through a composite structure such as a wall or roof is by multiplying the "conductance" of the wall by the area and the temperature difference. The conductance of the particular structure is found by referring to standard tables. Since these tables are rather voluminous, Table VIII has been provided as a compact set of data by which the characteristic of any structure may be calculated by simple addition.

The figures given are the resistances of the various layers through which the heat must pass; addition gives the total resistance of the wall. Note that the

TABLE IV.—DOOR CORRECTION FACTORS

(Multipliers for Table III—Double Glass)

Actual Thickness of Door, Inches	Multiplying Factor
25/32	1.5
1 1/16	1.3
1 5/16	1.2
1 3/8	1.1
1 5/8	1.0
2 1/8	0.85
2 5/8	0.75

TABLE V.—WINDOW CORRECTION FACTORS

(Multipliers for Table I—Weatherstripped, Poor)

Type	Multiplier
Double-hung metal	2
Same, weatherstripped	1
Industrial, pivoted, metal	5
Residential metal casement	1.5

TABLE VI.—WALL INFILTRATION

Structure	Infiltration With 15 M.P.H. Wind	
	Cu. Ft. Per Sq. Ft. Per Hr.	B.t.u. Per Sq. Ft. Per Hr. At 70F Diff.
Ordinary brick wall	7.85	10
Hard brick, cement mortar	2.6	3.3
Brick wall, plastered	0.07	0.09
Frame wall, plastered	0.13	0.165
Frame wall, unfinished	85	108
16" shingles on 1 x 4 lath	70	89
24" shingles on 1 x 6 lath	125	159

TABLE VII.—WIND CORRECTION FACTORS

Wind, M.P.H.	Factor
3	0.1
4	0.15
5	0.2
6	0.3
8	0.45
10	0.60
12	0.75
14	0.90
16	1.10
18	1.25
20	1.40
25	1.80

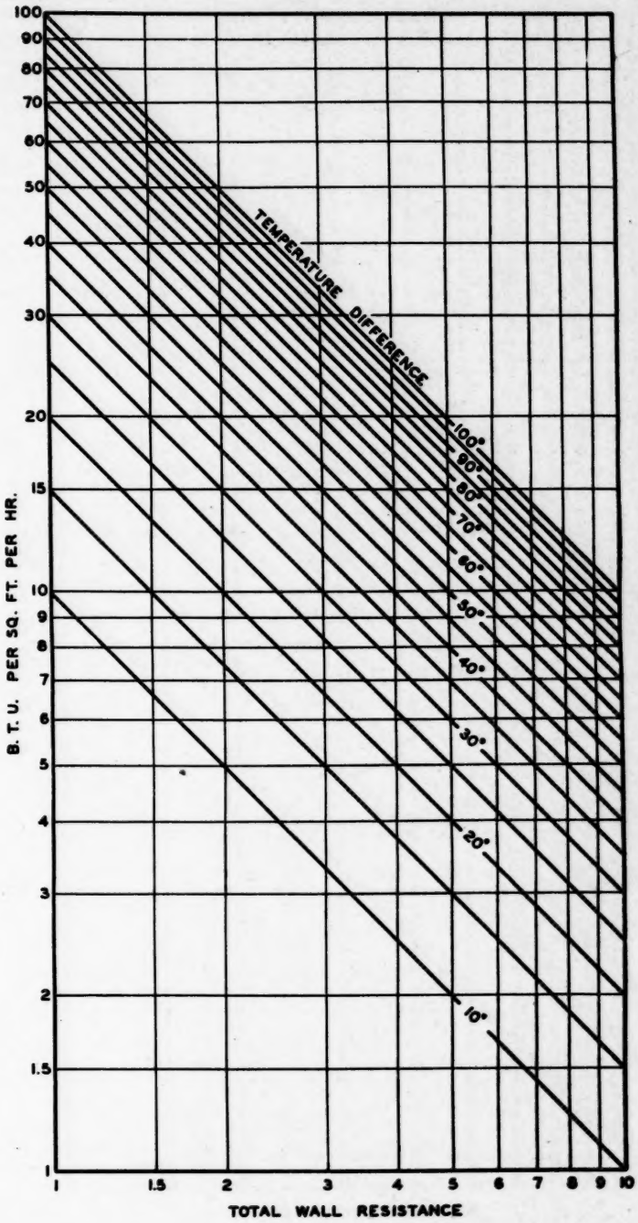


Fig. 1. Heat loss through walls and ceilings

outside surface has a lower resistance than the inside, because of the effect of wind. In the case of glass blocks, the figure given includes both surfaces, no addition being necessary.

The heat loss per hour through the wall may now be found by multiplying together the wall area and the temperature difference, and dividing by the resistance. To simplify this operation, the chart in Fig. 1 has been prepared, giving the heat loss per square foot for any wall resistance and temperature difference. It may be more convenient to assume a 70F temperature difference even though this is not the case. Several items calculated on the 70° basis may be lumped together and the adjustment made for the total.

Unheated Spaces

Ordinarily, some temperature such as 30F is assumed as existing in an unheated space, but this may often lead to considerable error. The accompanying chart, Fig. 2, has been designed to minimize the work involved in making an exact calculation. The procedure is, briefly:—

1. Calculate the heat loss through the entire heated wall adjoining the unheated space, assuming a temperature difference of 70F across this wall.
2. Calculate the heat loss through the exposed walls surrounding the unheated space, assuming the *same* temperature difference.
3. Enter the chart at the bottom with the calculated loss through the heated wall, and at the left with the calculated loss through the exposed wall. At the intersection, the diagonal line shows the *actual* temperature drop through the heated wall. Note that in using the chart, the decimal points may be moved the same number of places to the left for both heat loss figures. If one figure is more than ten times the other, the chart cannot be used, and it will be evident that the actual

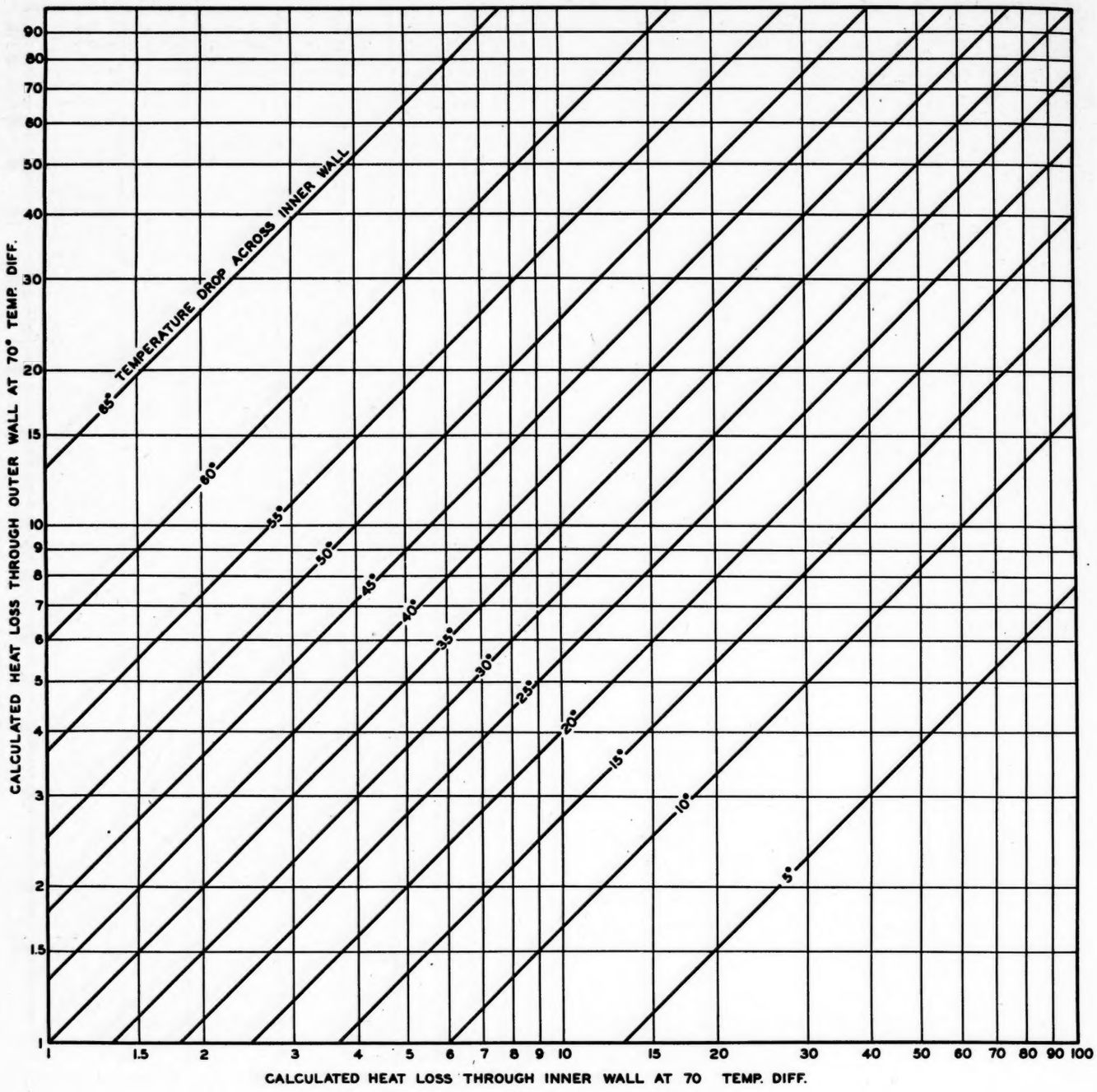


Fig. 2. Heat loss through unheated spaces

temperature drop will be close to zero or to 70F. In some cases, it may be necessary to divide both figures by 2 or some other simple number in order to obtain the same number of figures before the decimal point. At any rate, the determining factor is the ratio between the two calculated heat losses.

4. Adjust the heat loss calculated in step 1 to the actual temperature difference found by the chart. This may be conveniently done by using Fig. 1. Enter at the left with the calculated heat loss of the inner wall at 70F difference, ignoring the decimal point. Proceed to the right to the 70F line, then vertically downward to the line corresponding to the actual temperature drop, then left to the scale and read the actual loss. The decimal point will be the same as before, unless the horizontal line in the middle of the chart (at 10) has been crossed, in which case there will be one less figure before the decimal.

Losses Through Floors

No accurate data are available for estimating the heat loss through floors placed directly on the ground or over an unexcavated area with a dead air space between. Some designers assume a 5F temperature drop through the floor structure in such cases, and if the floor is of fairly large extent, this seems to be a safe figure to use. Normally, the loss at this point is small compared with the other losses of the building, so the final result can hardly be in error by more than a rather small percentage.

TABLE VIII.—HEAT RESISTANCES

(Resistance Is The Reciprocal Of The Conductivity Which Should Not Be Confused With Transmission Coefficient)*

Outer Walls and Roof	Heat Resistance
OUTSIDE SURFACE	0.17
MASONRY CONSTRUCTION	
Brick, face, 4 in.	0.44
Brick, common, 4 in.	0.80
Cinder block, 8 in.	1.61
Cinder block, 12 in.	1.96
Concrete, ordinary	0.08 (per in.)
Concrete, gypsum fiber	0.60 (per in.)
Concrete block, 8 in.	1.00
Concrete block, 12 in.	1.25
Stone; stucco; tile; terrazo	0.08 (per in.)
Tile, hollow clay, 4 in.	1.00
Tile, hollow clay, 6 in.	1.57
Tile, hollow clay, 8 in.	1.67
Tile, hollow clay, 10 in.	1.72
Tile, hollow clay, 12 in.	2.50
Tile, hollow clay, 16 in.	3.23
Tile, hollow gypsum, 4 in.	2.18
Glass block, 4 in.†	1.67
FRAME CONSTRUCTION	
Sheathing, fir, 1 in.	1.41
Siding, pine, 1 in.	1.18
Fiber insulating board	3.00 (per in.)
Gypsum board, ½ in.	0.38
Asphalt composition roofing, ordinary	0.15
Same, 3/8 in.	0.28
Same, bitumen and felt, slag surface	0.75
Shingles, asbestos	0.17
Shingles, asphalt	0.15
Shingles, slate	0.10
Shingles, wood	0.78
AIR SPACES AND INSULATION IN WALLS	
Ordinary air space, over ¾ in.	0.91
Same, faced 1 side with aluminum foil	2.17
Same, faced 2 sides with aluminum foil	2.44
Typical blanket insulation	3.70 (per in.)
Loose or batt insulation	
Mineral or glass wool	3.70 (per in.)
Regranulated cork	3.22 (per in.)
INSIDE WALLS, CEILINGS	
Plaster	
Cement plaster	0.13 (per in.)
Gypsum plaster	0.30 (per in.)
Metal lath and lime plaster, ¾ in.	0.23
Wood lath and lime plaster, ¾ in.	0.40
Plaster board (gypsum), ½ in.	0.38
Plaster board (gypsum), 3/8 in.	0.27
Cork board	3.33 (per in.)
Fiber insulating board	3.03 (per in.)
Plywood, fir, ¼ in.	0.45
INNER SURFACE	0.61
FLOORS	
Yellow pine flooring, 25/32 in.	0.95
Maple or oak flooring, 25/32 in.	0.77
Maple or oak flooring, 3/8 in.	0.37
Battleship linoleum, ¼ in.	0.74
Concrete	0.08 (per in.)

*Where the resistance is given on a per inch basis, multiply the thickness of the material in inches by the resistance to obtain the total resistance. †Includes resistance of inner and outer surface.

Losses through the portions of basement walls which lie below ground level are based upon the same assumption.

However, if the structure is such that the outside air circulates freely under the floor, it must be treated in the same manner as a wall.

The case of a room over an unheated garage presents an interesting example. If the garage doors are kept closed, the loss through the floor of the upper room is calculated by the method given for heat loss through unheated spaces. However, if the door is likely to be left open for more than a few minutes at a time, it should be assumed that outdoor temperature prevails in the garage. In either case, the use of insulation in the floor is indicated so as to preserve a reasonable degree of comfort, although such observations are somewhat beyond the scope of this article.

TABLE IX.—AVERAGE TEMPERATURES OF AIR NEAR WALLS AND CEILINGS FOR VARIOUS CEILING HEIGHTS.

(Referred to 70F at 4 feet from the floor)

Ceiling Height, Ft.	Aver. Temp. at Wall, F.	Aver. Temp. at Ceiling, F.
8	70.0	75.6
9	70.7	77.0
10	71.4	78.4
12	72.8	81.2
14	74.2	84.0
16	75.6	86.8
18	77.0	89.6
20	78.4	92.4

Adjustment for Temperature Difference

To avoid undue complication, many of the tables and charts have been constructed on the basis of 70° temperature difference. Because the design temperature difference may be some other figure, an adjustment must be made, and a similar adjustment is necessary to take into account the effect of high ceilings. These adjustments need not necessarily be made for each component part of the structure, as they may be grouped for convenience and the correction made in one operation.

In designing an entire heating system, or even in checking to determine whether each room has its proper share of radiation or leader pipe area, the rooms must be calculated individually, but for the simple purpose of determining the total heating load, the corrections may be made in a lump for the entire building.

High ceilings introduce a slightly more complicated situation. Here the average temperature drop through the walls is not the same as that through the ceilings, in fact if the ceiling heights are different on various floors, it becomes still more difficult to correct the calculation for the entire building at once.

Fortunately, the corrections are so quickly and easily made that it makes little difference. The method is identical to that used in connection with the calculation of the heat loss through unheated spaces. Entering Fig. 1 on the left with the heat loss on a 70F basis, proceed to the right to the 70F line, then vertically upward or downward to the actual temperature difference. The corrections for wall height and design temperature difference may be made in one operation; for example, suppose a certain room to have a calculated wall transmission loss (including window transmission and infiltration) of 5000 B.t.u. per hour, at 70F temperature difference. Suppose the ceiling height to be 12 feet, the corresponding average temperature on the inside of the wall is 72.8F, from Table IX. Suppose also that it is necessary to heat this room only to 60F when the outdoor temperature is zero.

Starting at "50" on the left-hand scale, proceed to the right to the 70F line, then upward to the (estimated) 73F line, then horizontally to the 70F line and downward to the 60F line. Coming back to the left-hand scale, the result is found to be 4400, the decimal point being placed by inspection.

Economical Dust Control with Low Resistance Exhaust Systems

By F. F. KRAVATH

Almost anyone can design an exhaust system that will work in a fashion. To design an exhaust system that will operate efficiently and economically requires skill and expert knowledge of the problem. The accompanying article is the first of a series of six articles dealing with economical dust control with low resistance exhaust systems. The author will conclude the series with the design of a sample system.

PART 1—ANALYSIS OF THE PROBLEM

AN exhaust system is a pretty expensive piece of apparatus, both to buy and maintain. It is made up of a number of elements each of which requires much study for its proper application, and the combination of which requires even more for true complementation and ultimate efficient operation. It is not enough to provide an excellent hood for a dust or fume nuisance if the rest of the exhaust system is inefficiently laid out and incapable of providing a sufficient volume of airflow through the hood. Nor is it at all better to burden a well designed exhaust system, handling more than enough air, and perfectly balanced, with a poorly designed, wasteful-with-air exhaust hood. The failure of any single part of the exhaust system will generally result in the failure or the inefficient operation of the rest of the unit and since nothing less than perfect performance is acceptable in dust control, it is a safe bet that the money could have been spent to better advantage had the proper study been given to all the various elements.

An exhaust system is a tailor-made job and as such must be suited to the individual needs and conditions of the particular processes to be controlled, and fitted to the physical properties of the workroom in which it will be installed. Exhaust systems cannot successfully be taken from one location and used in another unless the conditions of layout and operation are identical, and this rarely happens. What will suit the needs of one manufacturer may be entirely wrong for another, even though they may manufacture the same product. The point is that the exhaust system should be thoroughly investigated and worked out, if not wholly, then at least in conjunction with the manufacturer's own engineering staff. By knowing completely the product to be handled and the physical characteristics of the plant, we can pursue further the subject of this article.

Analysis

Let us first analyze just what we are trying to do and then let us do it in the soundest, most efficient manner, using all the technical data available today, and then let us compare it with the ordinary exhaust system so prevalent today. Let us compare it on the basis of good dust control in the workroom for both systems, and then judge the relative first costs, costs of operation, and costs of maintenance which all go to make up the overall costs. First of all, and this is very definite, we want good dust or fume control in the workroom. We want the concentration of harmful, nonbeneficial, or at least foreign substances in the air to be well below the accepted safe standards as set forth by various medical and hygienic groups. Such a listing appears in Table 1.

Dependent on the foreign substances liberated, their chemical composition, their size, and their amount is the method of securing the primary purpose of dust control, namely clean breathing air. If the process is not a dirty one, and gives off only small quantities of dust, fumes, or gas, then it may be entirely satisfactory to employ a system of ventilation such as shown in Fig. 1a and 1b, and which is described below. Here, it is desired to maintain a constant upward draft of 50 f.p.m. This type is not recommended for toxic materials.

(1) Mechanical ventilators or their equivalent must be employed at the workroom ceiling.

(2) Outside air must be admitted to the workroom near the floor level and in any event well below the dispersion level of the gases, fumes, or dusts.

(3) The entering air should, if possible, engulf or surround each production unit, or should enter through grates in the floor, while enough ventilators should be used and the spacing should be such that no pocketing of the contaminated air occurs.

(4) Enough air should be moved so that the concentration of foreign material in the air at the breathing zone is well below the standards of Table 1. The breathing zone is the space which the operator, inspector, supervisor or regular maintenance crew traverse, from the floor level to a point at least a couple of feet higher than head level, regardless of how high routine may force that level to be. It is the air at the breathing zone that we are concerned with and this must be safe. If the quantity of material given off is relatively low, and if the velocity of dispersion is low or directed toward the ventilator, a system of workroom ventilation such as described may well be practical. If the amount of material given off becomes too great, then an

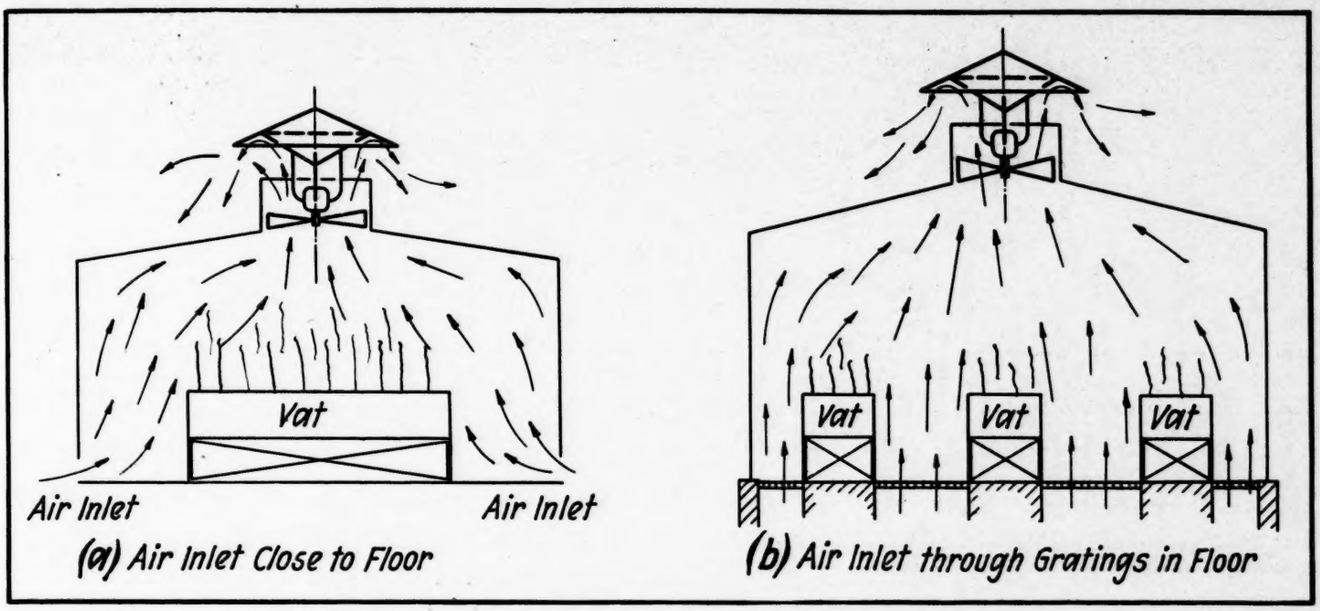


Fig. 1. Illustrating methods of ventilation, wherein the entire air contents are periodically exhausted in effort to keep air pollution low—suitable for non-toxic volatiles.

excessive amount of air will have to be removed to maintain clean atmospheric conditions resulting in a very cold workroom in winter, an intensely draughty workroom all year round, a higher than necessary power bill, much ventilating equipment, and more maintenance. The same conditions would result if, due to an unfavorably directed dispersion velocity, or due to the harmful nature of the contaminating substance, it were necessary again to move excessive volumes of air.

If the operation is hooded, and a stack of relatively large diameter is directly coupled to each hood, then, depending on the height of the stack, the number of bends or obstructions, and the difference in weight between the gases or fumes and that of air, satisfactory operation may well result through the use of natural draft ventilation alone. See Fig. 2. However, it must be stressed that the foregoing two methods are desirable only in handling non-toxic gases, fumes, or dusts given off in relatively small quantities, at no dispersion velocity, and with the aid of mechanical ventilators, favorable air inlets, hoods or natural ventilation and, in any event, the safe or desirable limits of air pollution must not be exceeded.

In most manufacturing processes in which gases, fumes, or dusts are formed they are emitted either at high dispersal velocities, in high volumes, in many directions or some combination of the foregoing, or they are toxic, heavier than air or both so that from every practical standpoint local exhaust ventilation including ductwork and power-driven exhausters are essential. The foregoing two cases cited and illustrated can not be classed as real exhaust problems, being more of the ventilating type of problem where it is desired to prevent foul, stale, or musty atmospheric conditions. In the real honest-to-goodness exhaust problem, we have far different conditions. Here we have material given off in quantity, much of it toxic, and much of it coming off at a high dispersal velocity. By building a hood that will enclose the process, the material is restrained from polluting the workroom air. If the process were entirely automatic it would be possible to secure good dust control by enclosing the process completely (provided all leaks could be eliminated); thus no exhaust system would be necessary. However, due to the fact that many processes require operators in constant or periodic attendance for purposes of inspection or manufacture, it is necessary that some portion of the hood be left open to accommodate the operation. This, in many cases, is quite a considerable portion of the hood. Since most operations antedate the hoods, the operators, used to a great deal of freedom, are quite loath to part with any portion of it. Here, great tact as well as perseverance are necessary in order to work out a hood which is satisfactory to the operator as well as efficient in performance.*

It must be remembered, that while we may endeavor so to design the hood that material will be deposited within it, still there is always a certain amount of fine dust which will come out of any opening unless there is a sufficient flow of air into the hood through the opening or openings. Dependent on the individual condition is the velocity of the air stream necessary to restrain the material from escaping. While in some fume hoods, a face velocity of 50 f.p.m. is sufficient to control the hazard, in the case of some high speed grinders, it

*Since it is necessary, in any case, to know the operator's exact movements in order to provide the proper clearances, it is invariably best to draw the operator out. get him interested in the problem, and listen to his ideas if offered. Never make the mistake of telling the operator his ideas are no good and, above all, never ask the operator if he has any ideas on the subject. In the first case, he will never be satisfied with the hood, while in the second case, he will think you have no ideas of your own and are trying to exploit his brains to your own benefit. These comments are drawn from considerable experience in hood design, the most delicate part of the exhaust system.

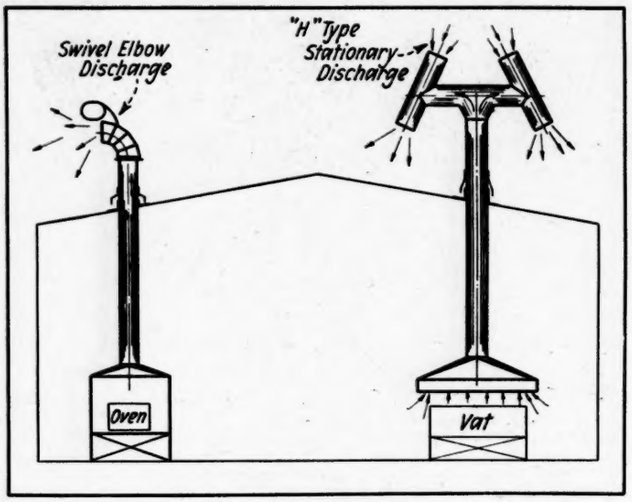


Fig. 2. Illustrating natural draft ventilation suitable for non-toxic volatiles where processes are well hooded or enclosed. "H" type of discharge illustrates inspirational effect of gust sweeping down stack legs.

TABLE 1—TOXICITY OF GASES AND FUMES†

Vapor, Gas or Dust	Maximum Parts of Contaminates Per Million Parts of Air Allowable For Prolonged Exposure
Ammonia	100
Amyl Acetate	1,000
Butyl Acetate	1,000
Aniline	7
Arsine	20
Benzol	100
Butanol	100
Carbon Bisulphide	1
Carbon Monoxide	100
Carbon Tetrachloride	1,000
Chlorine	1
Chloroform	200
Ethyl Bromide	1,700
Ethyl Chloride	20,000
Hydrochloric Acid gas	10
Hydrocyanic Acid	20
Hydrofluoric Acid gas	3
Hydrogen Sulphide	10
Mercury	40
Methyl Alcohol	100
Methyl Bromide	150
Methyl Chloride	750
Naphtha	5,000
Nitrobenzol	1
Nitrous Oxides	35
Phosgene	1
Phosphene	20
Sulphur Dioxide	10
Tetrachlorethane	10
Trichlorethylene	250
Toluene and Xylene	150
Turpentine	700
Ferric or Zinc Oxide*	5.00*
Chromic Acid	0.10*
Lead Oxide Dust	0.15*
Silica Dust	1.00*‡
Asbestos Dust	4,000,000ˣ
Any Dust	20,000,000ˣ

†Data secured from Industrial Dust, by Drinker & Hatch; General Orders, etc., Industrial Commission of Wisconsin; Public Health Bulletin No. 217; Bureau of Mines Bulletin No. 400; A.S.H.V.E. Guide, 1939; and Diseases of Occupational & Vocational Hygiene, by Kober & Hanson.

*Maximum allowable contamination in milligrams per cubic meter of air.

‡Or 5,000,000 particles per cu. ft. of air, particles being less than 10 microns in longest dimension and dust being 35% or more free silica.

ˣAllowable number of particles per cubic foot of air.

is necessary that hood face velocities be as high as 1000 f.p.m. However, it must be borne in mind that velocity—not volume—controls the hazard. Volume is only gotten from the product of the velocity and hood opening. Obviously, if a hood face velocity of 100 f.p.m. for a certain hood is sufficient to control the hazard, it is foolish to use 500 f.p.m. Or if the hood opening, through careful design, can be reduced from 4 sq. ft. to 2 sq. ft., it is certainly worth the trouble. In the instance cited, a minimum of 200 c.f.m. will be moved as against an original 2000 c.f.m. Whether there are twenty machines concerned or two, the saving is still worthwhile. Here, then, are the first two maxims on economy:

(1) On the basis of good dust control, *keep the hood face velocity at its lowest safe figure.*

(2) *Keep the hood opening as small as possible.*

While the first thought was to keep the material from escaping to the room atmosphere, it must now be decided whether all the material caught within the hood is to be transported to some collection center for filtering, whether the larger sized material will be dropped out in a cyclone or gravity filter after a short distance of conveying, or whether this will be done in a chip trap built into the hood or near it. Where the material is all fine and relatively light, it can all be conveyed to a central point utilizing a low pipe velocity, such as 3000 f.p.m. Fumes, gases, and the finest dusts should never be conveyed at any higher speeds, since this is quite ample and it must be borne in mind that the resistance to flow in any pipe varies approximately as the square of the pipe velocity, while the power to maintain flow is directly proportional to the resistance. While it is always best to ascertain experimentally the correct conveying or pipe velocity, it may be gotten almost as closely from the following formula, as expressed by DallaValle:

$$V = 54{,}700 \left(\frac{S}{S+1}\right) p^{1/2}$$

where

V = Conveying velocity, in f.p.m. for vertical transport of material
S = Specific gravity of the material
p = Particle diameter in feet

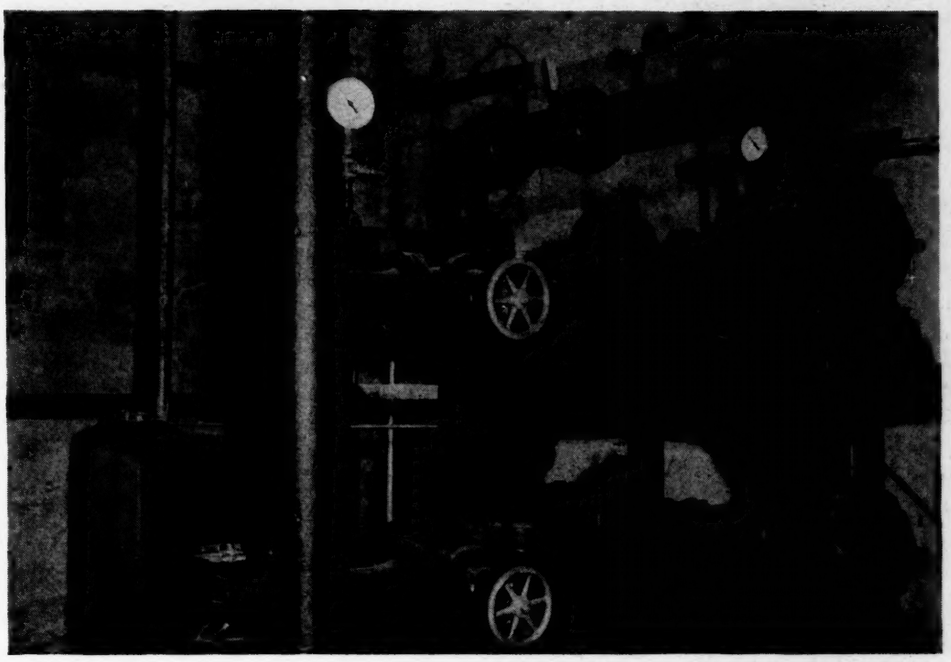
Having arrived at the proper conveying velocity, allow between 25% and 50% more for safety, so as to prevent any precipitation of material within the pipes since this would soon cause poor performance. Since the volume of air to be handled is determined at the hood, as already explained, the pipe size is gotten by dividing the volume by the pipe velocity, thus

$$Q = A\,V,$$

where

Q = Volume of airflow in c.f.m.,
A = Pipe cross-sectional area in sq. ft.,
V = Pipe or conveying velocity.

[*Next month Mr. Kravath will discuss hood and branch connections and the importance of efficient elbows.*]



Pump room in the Dixieland plant at Fort Knox showing hot water circulating pumps which serve 71 units of commissioned officers quarters. At left is an auxiliary driven motor pump for use in power failure emergencies.

Fort Knox Defense Housing Heated With Hot Water

During the next few years many engineers and contractors will be working on the heating of housing units for the rapidly expanding army. Although systems for heating these buildings are basically the same as for civil buildings, they differ in that they must be able to be quickly installed and meet the special conditions of military structures. In this article is described the systems installed at Fort Knox, Kentucky, which has been recently greatly expanded. All of the systems described were installed by the Works Projects Administration.

A great military establishment is being forged with all speed on a 35,000-acre tract at Fort Knox, Kentucky.

Up in the hills of Hardin County, Kentucky, nestling behind the high range of the Cumberlands, 600 miles from seaboard, the site of the nation's gold reserve also has been designated the principal training center for mechanized fighting forces, a center for specialized National Guard training, and a garrison post that will include infantry, air squadrons and other units.

The development of the Fort Knox reservation, which is peculiarly strategic since the erection of the gold fortress five years ago, is now a part of America's defense program. The site, for those who know the Cumberland range in the Alleghenies, is considered unassailable from the coastal lowlands and ideal for secret maneuvers.

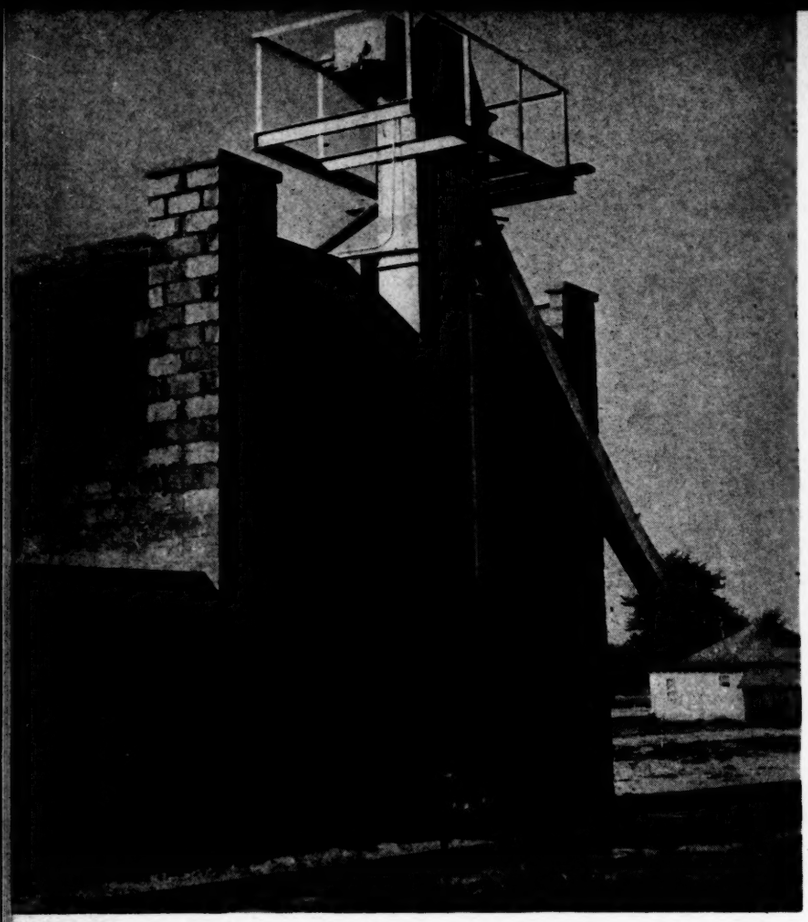
Construction work at this Fort has included a modern flying field, sheds to house 1,000 pieces of motorized equipment, roads, barracks, mess halls, tent floors, offices, streets, sewers, water mains, light conduits, gas mains and landscaping.

The modern flying field, known as Godman Airport, occupies 1440 acres. It extends one and one-half miles in either direction and has four 150 ft. wide runways, one a mile long, two 4,000 ft. long and one diagonal runway 3,500 ft. in length.

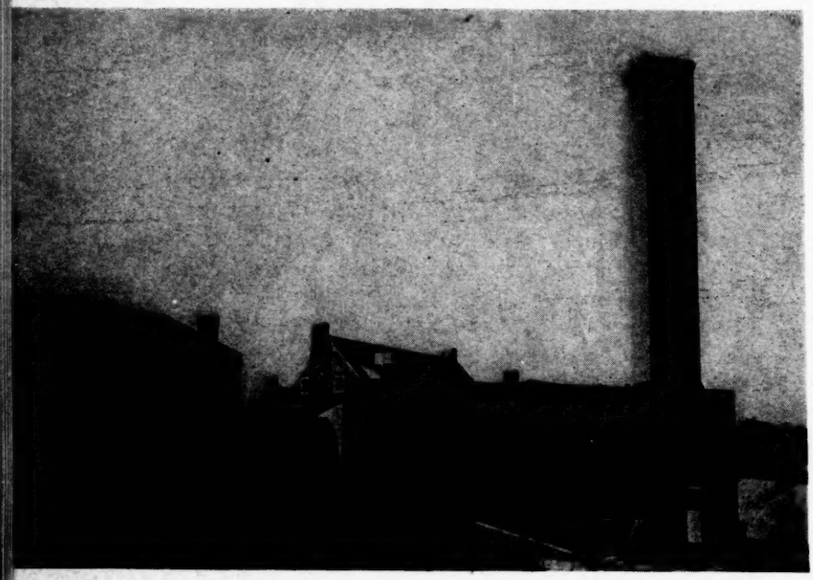
A large stone quarry and concrete block plant operated by the WPA furnished much of the material needed for the building. Among many other uses, the blocks have been used in the construction of basements under 46 sets of non-commissioned officers' quarters and 36 sets of officers' quarters; construction of 80 mess halls; 62 non-commissioned officers' quarters; as foundational material for motorized equipment sheds; and for 1,000 manhole covers.

In a section set aside for the National Guard training camp there are 2,400 concrete tent floors, 16 by 16 feet, each wired for electricity; 44 concrete block mess halls for 100 men, and washrooms of the same material. The tents will care for eight men, and it will be possible to house 19,200 troops in an hour. The objective is to provide the most permanent and sanitary type of quarters possible. Adjoining quarters, assigned to the Sixth Infantry, are of similar construction.

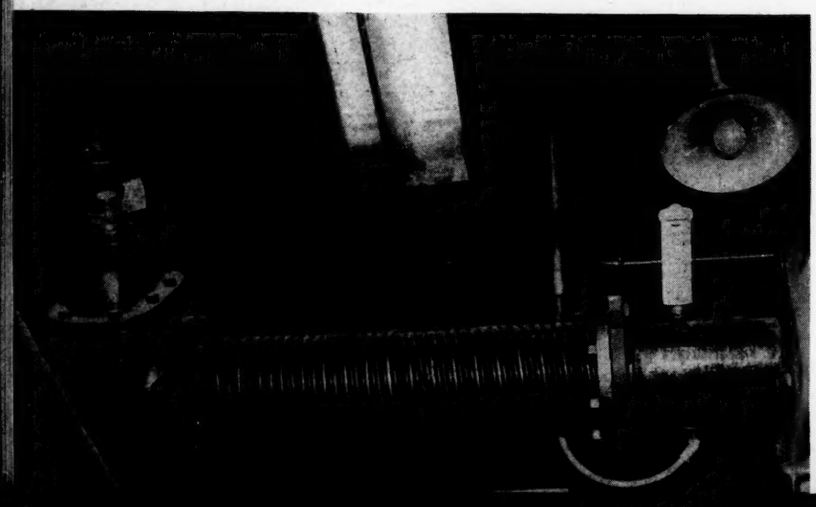
Heating, a major factor in the smooth operation of the huge army post at Fort Knox, Kentucky, has un-



Heating plant No. 278 which is equipped with a bucket elevator capable of handling a car-load of coal at a time.



(Above) Dixieland heating plant which heats 71 homes of the type shown at the left. (Below) Corrugated steel pipe connection used between boilers and main feeder line.



dergone since the advent of the preparedness program drastic revision from standard permanent circulating hot water systems to various other types, quickly available in the emergency.

After the emergency has passed, the Fort Knox establishment probably will return to its standard type of construction to complete a vast system which already is well advanced.

Among the permanent building projects assigned to the WPA at Fort Knox was the construction of five modern central heating plants, together with many other plants for single buildings. All of these heating plants at Fort Knox are of the circulating hot water type.

Dixieland Plant

The first of these central heating plants was placed in operation early in the fall of 1939 and is known as the Dixieland Plant. Ths plant furnishes heat to 71 sets of permanent brick commissioned officers' quarters from three welded boilers with a three-inch asbestos covering. Each boiler is rated at 29,960 sq. ft. of hot water radiator surface. The boilers are fired by three bin-fed stokers, fed from the rear of the boiler to conserve space. The coal bins from which these boilers are fed are of two-car capacity.

At all times one of these boilers is held in reserve for an emergency. Each is equipped with automatic pop safety valves. The circulation system is handled by electrically-driven centrifugal pumps. In the pump room also is installed an auxiliary gasoline-driven pump for use in case of power failure.

In this connection it is interesting to know that last winter this plant suffered a seven-hour night power failure when the thermometer registered between —12 and —15F. However, water in the radiators remained sufficiently warm to prevent any damage or discomfort during the time of the power failure.

Recording instruments are used in the Dixieland Plant as a part of its control system. On a weekly chart the inside and outside temperatures are recorded, together with the temperature of the outgoing and incoming water in the circulating system.

Each officer's home, served by the Dixieland Plant, is equipped with an individual thermostatically controlled, electric motor-driven valve for heat control. This installation maintains constant temperatures and governs an automatic cut-off valve which diverts the hot water from the house back into the main circulating system when a desired temperature is reached. It automatically cuts on with a drop in temperature.

To install the circulating system for the Dixieland plant, there was laid 10,138 ft. of pipe ranging from 2 in. to 8 in. Joints in all of the pipe, excepting the 2 in. size, are welded. Eight inch pipe was used for lines leaving the plant, while 5 and 4 in. lines were used for branch headers. In carrying the system into the 71 houses, pipe from 3½ in. to 2 in. in size was used. In the entire system, 36,216 sq. ft. of cast iron radiators were installed.

Constant temperatures are imperative in the handling of the immense quantities of food stuffs necessary for use at Fort Knox, especially in those warehouses where canned goods are stored.

To supply quartermasters' warehouse numbers 1, 3 and 8, and also a series of quartermasters' offices, there was constructed a concrete block central heating plant known as building number 278.

The heating equipment in this plant was originally installed at the quartermasters' depot at Jeffersonville, Indiana, but was dismantled and moved to Fort Knox.

The plant contains two cast iron, down-draft sectional boilers with water grates. Hot water is forced through a circulating system by means of centrifugal pumps.

The plant is hand fired from a three-car capacity coal bin equipped with a bucket elevator capable of handling a car-load of coal at a time. This elevator also is used to load coal direct from car to trucks for other post sites.

Post Headquarters

Due to the rapid expansion of Fort Knox under the present emergency program, a new post headquarters became a necessity. Two old barracks buildings were renovated and joined for this purpose and a modern heating plant was built to replace previous cast iron stove heat.

The plant now serving post headquarters is known as building number Q53. It is hand fired and equipped with a cast iron sectional boiler for forced hot water heating. The boiler is of the cast iron sectional type and is installed at ground level.

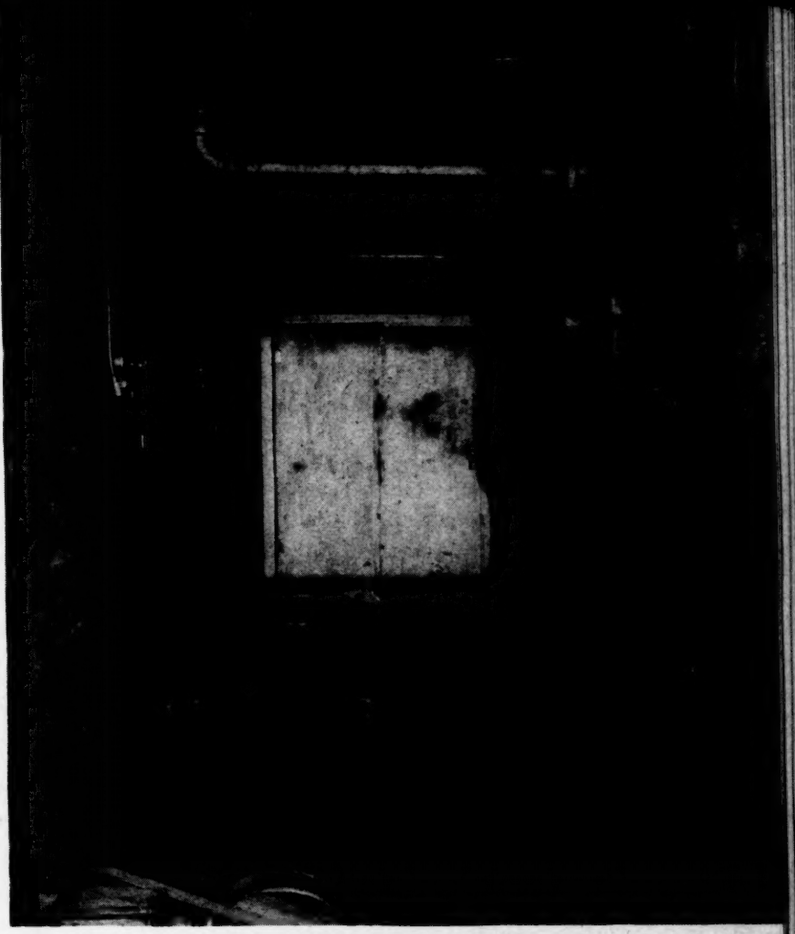
Ground level setting of boilers of this type has been made possible by the development of economical electrically-driven circulating pumps with low heads and small motors. Motors for these pumps are mounted in rubber to reduce noise and vibration. To insure constant water pressure the plant is equipped with a system of automatic feeders.

Included in the project was the construction of another hot water heating system to supply three houses that were rehabilitated by WPA.

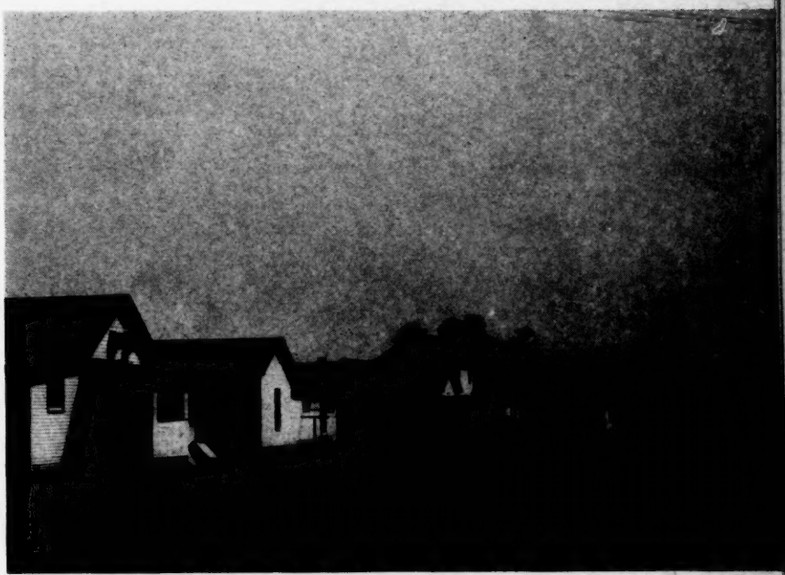
This new plant replaces an old boiler room with a gravity type steam boiler heating two old houses and uses less fuel to heat three houses than the old plant used for two.

The new plant is known as building number CC18, and contains a new sectional boiler with the necessary coal bins and electrical circulating system. The plant is hand fired.

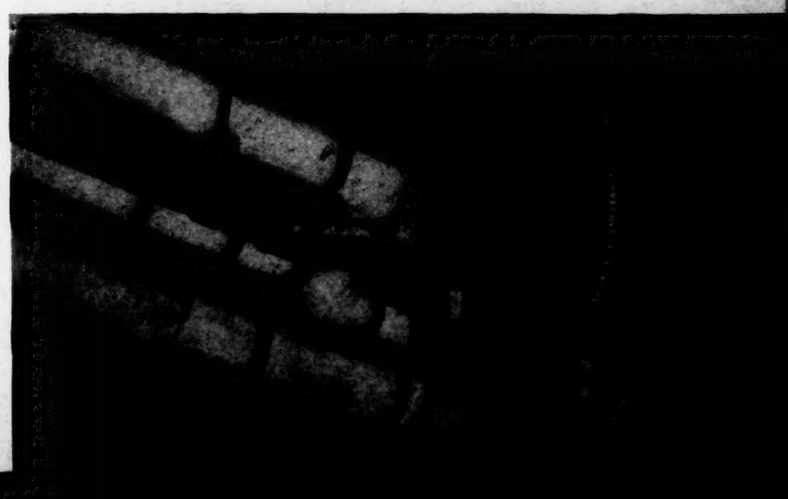
Six old residences were moved several blocks down the street and rehabilitated for non-commissioned officers' quarters. To supply these residences with heat, there was constructed a concrete block hot water central heating plant of the same type previously described. The plant is hand fired and replaces six individual heating units. In fifty or more permanent type buildings erected at Fort Knox, individual hot water heating plants of smaller capacity have been installed. All, however, follow the same designs of the larger plants.



Three bin feed stokers which feed the boilers from a two car capacity bin at the rear.



A small central hot water heating plant which serves the six homes of non-commissioned officers shown above. (Below) Electrically driven thermostatically controlled valve in commissioned officers homes.



Notes on Installing and Servicing Bituminous Coal Stokers

By WALTER KNOX†

The author, experienced in the testing development and production of stokers, gives here some pertinent and very practical advice on the installation, servicing and operation of domestic bituminous underfeed stokers. The points covered by the author are abstracted from an address made at the fourth annual convention of the Virginia Retail Coal Merchants Association.

RIGHT at the present I am trying to learn more about the behavior of various coals in stokers. For the past few years I have had the privilege of working with stoker companies that were doing intensive and fundamental research on the best method of putting coal into a boiler and burning it. Perhaps some of the things we learned will be of value to you.

The first thing I want to mention is something of which you have all heard and probably generally find in stoker installations. That is a barometric damper. Do not fail to recommend one. An automatic damper is a fuel saver, it promotes a more uniform depth of fuel bed, and all the evidence shows it tends to help in stubborn cases of fires going out. There should be one on every job. Set it so the overfire draft is three or four hundredths of an inch when the stoker is running.

Troublesome clinkering and certain other difficulties may result from an incorrectly made hearth or from unnecessary pitting of the stoker.

A question often asked is, should the hearth be level, sloped toward the retort, or sloped away from it? Make the hearth flat. If you must slope it, slope it away from the retort. Never make a dish pan effect around the retort. Make the hearth flat and ordinarily get as much space as possible for clinkers. Only in cases of exceptionally large or long fire boxes it is necessary to build bridge walls to confine the fire. It is desirable to have five or six inches of flat clinker space (approximately one-half the width of the retort) around the ordinary small domestic size retort.

This question of the hearth brings up the questions of protection of the water legs of the boiler and that leads to the question of a pit. I never like to see a stoker put in a pit unless it is absolutely necessary and most of the time it is not necessary. And this is especially true if a low-volatile coal is used.

If a retort is in a pit it has brick walls around it. These walls absorb no heat for delivery to the heating system, in fact they reflect heat down on the retort and shorten its life. On the other hand if a retort is up at grate level the boiler walls are absorbing heat as they are meant to do. Time and time again it has been shown that raising a stoker out of a pit increases the efficiency. Moreover you do not have to protect the water legs and there is more space for clinkers. Also generally it is easier to remove clinkers.

The idea that so much head room is required is an erroneous carry-over from big boiler installations on process or power production work. What we have been saying is, this stoker feeds so many pounds of coal an hour and for every pound burned we have to have a minimum amount of combustion space. That is true in a big plant running continuously at maximum rating for considerable periods. For small home heating installations it is not true. Herein lies the fallacy:

The small heating unit does not run continuously for any great length of time. Possibly you will say, "But when it is running it is running at full speed." No! It is putting coal in at full speed, let's say 30 pounds an hour, but it is not burning at that rate.

Here is what happens then in the case of this assumed 30 pound stoker. A demand for heat causes it to put in coal at the rate of 30 pounds per hour for a certain length of time, one-half hour, or one hour, or whatever the requirements of the house demand. During that time only part of the coal is being burned. We have all seen fuel beds built up during *on* periods. Then the stoker stops, and during this *off* period the remaining coal continues to burn by natural draft, keeping the boilers warm, some of the basement pipes warm, the house floors warm, and here we have the most satisfying advantage of stokers as they are being built, no cold 70's, no cold floors, a readiness for instant response to the thermostat because the boiler and pipes are warm.

To get back to the question of headroom and stoker settings we see more clearly now why head rooms based on the maximum coal feed of the stoker are misleading, we see why that if a boiler was large enough when hand-fired, burning coal on the grates, then it is generally best to put the hearth at grate level, giving maximum heat absorption by the boiler water walls, maximum clinker space, and maximum ease in removing clinkers.

The question arises, "Is this an efficient and satisfactory way of burning coal?" The answer is, "Yes, it is." An underfeed stoker has the ability (even though feeding more coal than it is burning) to burn what is being burned quite efficiently. The ability varies with different types of coal and also the condition cannot be carried on indefinitely. If the period of continuous running is too long, or the air is too low, there may be reached a condition when smoke or a percentage of CO is formed. In that case we turn on more

†Director, Koppers Stoker Coal Research Laboratory.

air—the fuel bed has grown too high. So in practical application we say to adjust the air so that under the conditions of operation, a fairly deep fuel bed is maintained, yet a bright clear flame is attained. That is the practical answer and theoretically we find by analysis it is correct.

Mechanical Features

I now want to take up some mechanical features of stokers that affect burning. While I had said that the day is past when the stoker man always blames the coal and the coal man always blames the stoker, yet there are older stokers still in use that do not have the refinements of the present best models. Also it is always possible for now and then a mis-assembly to take place or for something to happen in use. Certainly there are some times when a coal that burns satisfactorily in other stokers gives trouble in a certain stoker.

Have you ever seen a lifeless, smoky fire? This condition, and also uneven burning, and burning out of retorts may be caused, especially in older stokers, by improper fan and tuyere design. Some of the older stokers did not have fans of sufficient capacity to deliver air against a relatively high resistance, that is, a high fuel bed. I have mentioned, what you have all seen, the way a fuel bed builds up as a stoker answers a demand for heat. As that fuel bed builds higher and higher the air must penetrate the bed if efficient combustion is to take place. We do not need a great quantity of air, but it must get up through and over the fuel bed. So we need a fan that will deliver the air, not in great quantities, but in correct quantities against a resistance. The type of fan that will do that is one with a good diameter rather than width. Some of the old fans were too wide and not of sufficient diameter. It is diameter that gives a fan ability to develop static pressure.

Stokers can and are being made that consume very little power in worming in coal. Such stokers produce only a moderate amount of crushing. Let's see how it is done in order that you can correct the trouble in some orphan machine you may have or at least diagnose the difficulty if occasion should arise.

In the first place the coal should not be packed in the coal tube, rather it should be flowing rather loosely. To accomplish this one common step is to meter the coal at the spike trap, or entrance to the coal tube. This entrance point then is made the smallest diameter of any place in the coal tube. Often a service man who finds a stoker working hard will ease up on the spike trap, lift it up some, thinking to ease up on the load. What he is doing is making the load worse by making a funnel effect in which the coal is going in the large end and trying to get out the small end. On the contrary the spike trap should be kept down and there should be a tight fit around the feed worm at the spike trap.

As a rule this alone is not sufficient. As a corollary in metering the coal a practice of some manufacturers is to make the pitch of the worm (distance between flights) less in the hopper than in the coal tube. By this method the coal is fed into the entrance of the tube rather tightly and then it loosens up when it comes to the area of greater distance between flights.

Another method used by some companies is to reduce the diameter of the feed worm inside the hopper. This is very effective. The diameter may be reduced in one step or by a gradual taper. One advantage of this method of reducing the diameter is that it also reduces the churning and agitation of the coal and thereby reduces crushing, segregation and noise.

Now of course noise may come from coal of too large a size for the particular stoker. A sized coal with a small top size feeds easily and uniformly and with no noise and little segregation. This is ideal.

Packing and Crushing

The question comes up of what can you do if a stoker feeds too much coal into the coal tube, causing packing and crushing. In the first place you can see that the spike trap is down as close as possible to the feed screw. Another thing that often helps a lot, if the diameter at the entrance to the coal tube is too great, is to put a half round of sheet metal, or even heavier material, on the bottom of the hopper base. Also it is not difficult to cut down the diameter of a feed screw with a torch. Sometimes it is necessary to use all the possible remedies to achieve the desired results.

There is only one possible danger in connection with such remedies that loosen up the coal. That danger is that you make it too easy for smoke to come back through the coal tube. With low-volatile free-flowing, properly sized stoker coal the danger of smoke-back is very small. However, it might be well to see how smoke-back is controlled.

Practically all manufacturers put holes in the top of the coal tube up near the retort and put in an air pressure through these holes. If a high fuel bed exists and air carrying coal gas tries to come back from the retort it is met and equalized by air pressure from the holes in the top of the tube. Now of course there must be some resistance to air in the coal tube itself so the practice is to slightly pack the coal under the spike trap for a short distance. This is done in the case of variable pitch feed worms by carrying four or five inches of the smaller pitch flights into the coal tube. In the case of a reduced diameter of worm in the hopper, the full diameter is retained for three or four inches nearest the entrance to the coal tube.

At the present time most manufacturers seal their hoppers on the domestic sizes and some even put air pressure from the fan into the hopper itself. At any rate the holes in the coal tube should be inspected in case of trouble and if closed they must be cleaned out.

I have been talking about service problems about which the stoker manufacturer possibly should be held responsible. However, there is one thing that coal men should be responsible for and it is very important, that is to tell people how to clean a fire and insist that they do it.

It is only a short easy job to properly clean a fire, yet too often it is not done and the result of not doing it causes many service calls. Tell that home owner to take the clinkers out regularly, take them all out, and then to rake the remaining ashes into the space from which the clinker was removed. Take out all the biggest clinkers and rake the rest of the ash and clinker toward the retort.

ABSTRACTS of Current Papers, Books and Pamphlets

Dust Collectors

In an earlier paper on this subject, the authors' presented data on a twelve-inch glass model cyclone which showed the effect of cyclone proportions on the fluid friction loss and on the flow pattern within the cyclone. In these tests the cyclone inlet contained a variable inlet vane which projected half-way into and across the cyclone cylinder.

The tests reported in this paper were conducted to determine the validity of a claim made by John L. Alden in HEATING & VENTILATING, that the use of a deflector vane at the cyclone inlet reduced the cyclone pressure drop by more than 50%. As a result of their studies the authors observed that the total friction loss in the system without the deflector vane is 5.47 in. of water whereas it was previously computed to be 3.08 in. of water with the deflector vane, an appreciable difference when the fan performance must be specified.

Thus, they concluded that Alden was correct in that an inlet vane of proper design will substantially reduce the pressure drop through a cyclone collector. No quantitative results are available as to the effect of such a deflector vane on dust collection efficiency but where pressure drop is an important factor, a cyclone of the type suggested in the earlier paper combining an inlet deflector vane with exit vanes of suitable design, should give efficient collection at a low pressure drop.

[*"Flow Pattern and Pressure Drop in Cyclone Dust Collectors," by C. B. Shepherd and C. E. Lapple. Published in Industrial and Engineering Chemistry, Room 706, Mills Building, Washington, D. C., September, 1940, Volume 32, No. 9, pages 1246 to 1248. Price, single copies, 80 cents.*]

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Ceiling Insulation

This investigation deals with the effect of insulation in limiting the temperature rise of the rooms in the upper story of a building with a flat roof. For test purposes a structure was built on the roof of one of the buildings at the National Bureau of Standards. This structure has a flat (slightly sloping) roof, below which is a low attic separated from the room below by a ceiling of plasterboard. Various types of insulation were installed just above this ceiling to insulate it from the attic space. The temperature of the ceiling in the room below was measured, and the effect of the insulation in limiting the rise in the temperature of the ceiling was determined.

The results show that a room that has nothing but a plastered ceiling between the occupants and the roof boards will not be comfortable on hot summer days, especially if the attic is not ventilated. If there is no attic floor, a couple of inches of blanket or fill insulation put on top of the ceiling, or a single layer of aluminum foil tacked over the tops of the joists, will greatly improve conditions. More than two inches of insulation will produce only slightly better results.

If the attic has a floor, about two-thirds of the improvement that would result from using insulation has already been attained. Placing additional insulation under such a floor may be rather expensive, and, although the room may be perceptibly more comfortable with the insulation, there may be doubt as to whether the gain was sufficient to warrant the trouble and expense.

[*"Effect of Ceiling Insulation Upon Summer Comfort. Building Materials and Structures, report BMS52, by Thomas D. Phillips. Published by the National Bureau of Standards. Paper cover; 8½ x 11 in.; 10 pages. Price, 10 cents. For sale by the Superintendent of Documents, Washington, D. C.*]

BRIEF REVIEWS

GAS FURNACE SELECTION. Gives information on the sizing, locating and installing of gas-fired floor furnaces in residential buildings. [*"The Gas Floor Furnace" by A. F. Rice. Published as an Interim Bulletin of the House Heating and Winter Air Conditioning Committee, American Gas Association, 420 Lexington Ave., New York. Also available as a reprint from Gas, April, 1940, published by Western Business Papers, Inc., 1709 W. Eighth St., Los Angeles, Cal.*]

BOILERS. A revised and enlarged edition of the IBR test and rating code for low pressure heating boilers. Includes information on the history and development of the code, and outlines in detail the procedure by which IBR ratings are obtained. [*"IBR Testing and Rating Code for Low Pressure Heating Boilers." Published by the Institute of Boiler and Radiator Manufacturers, 60 East 42nd Street, New York, N. Y. Price, 30 cents.*]

EDITOR'S NOTEBOOK. A service bulletin for employee publication editors. Presents information on various phases of publishing employee papers and magazines. [*"Post Convention Issue, A Bulletin for the Editor's Notebook," June, 1940. Published by the American Association of Industrial Editors, Inc., 1964 East Ninth Street, Cleveland, Ohio. 8½ x 11 in.; 24 pages.*]

SMALL HOUSE HEATING. A booklet illustrating seven methods of heating small houses. Systems illustrated include: one-pipe steam, two-pipe steam, two-pipe hot water gravity circulation, one-pipe hot water forced circulation, two-pipe hot water forced circulation, warm air gravity circulation and warm air forced circulation. Should be valuable to contractors in explaining types of heating systems available to small home owners. [*"Basic Ways to Heat the Small Home." Published by McDonnell & Miller, Wrigley Building, Chicago, Ill. 8½ x 11; 4 pages. Available on request.*]

WEATHER. A series of papers prepared in an attempt to get an objective appraisal of the value of methods of long-range weather forecasting. [*"Reports on Critical Studies of Methods of Long-Range Weather Forecasting." Supplement No. 39. Published in the Monthly Weather Review of the United States Department of Agriculture, Weather Bureau, Washington, D. C. Paper cover; 9½ x 12 in.; 130 pages. Available from the Superintendent of Documents, Washington, D. C. Price, 20c.*]

WELDING. A series of lessons which presents in a concise manner certain fundamental facts of welding which will enable welding operators to utilize welding processes successfully and economically. Book forms a basis of instruction in the Lincoln Arc Welding School. [*"Lessons in Arc Welding." Published by the Lincoln Electric Company, Cleveland, Ohio. Cloth bound; 6 x 9 in.; 138 pages. Price, 50 cents postpaid in the United States. Elsewhere, 75 cents.*]

SPRINKLER SYSTEMS. A pamphlet which offers to property owners and managers advice and suggestions on the care and maintenance of sprinkler systems. [*"Care and Maintenance of Sprinkler Systems." Published by the National Board of Fire Underwriters, 85 John Street, New York. Paper cover; 6 x 9 in.; 18 pages. Available on request.*]

PIPE NIPPLES. Commercial standards for pipe nipples made of brass, copper, steel and wrought iron. This is a recorded voluntary standard of the trade which is effective for new production from May 10, 1940. [*"Pipe Nipples; Brass, Copper, Steel and Wrought Iron. Commercial Standard CS5-40." Published by the National Bureau of Standards, United States Department of Commerce. Paper cover; 6 x 9 in.; 15 pages. For sale by the Superintendent of Documents, Washington, D. C. Price, 5 cents.*]

REFRIGERATION. This book has been written primarily to present a comprehensive picture of the field of commercial refrigeration and comfort cooling for the use of mechanics, salesmen, architects, and engineers. The section on air conditioning presents very superficial information which is available in a better form in other books. [*"Commercial Refrigeration and Comfort Cooling," by Samuel C. Moncher. Published by Nickerson & Collins Co., 435 No. Waller Ave., Chicago, Ill. Cloth bound; 6 x 9¼ in.; 109 pages. Price, $1.50.*]

HOUSING. A booklet designed to present a simple and authoritative discussion of basic housing facts and principles which are of concern to every community determined to clear its slums and to provide decent homes for families of low income. Contents should be of interest not only to those who are actively associated with the housing movement but also to the public at large. [*"Introduction to Housing, Facts and Principles," by Edith Elmer Wood. Published by the Federal Works Agency, United States Housing Authority. Paper cover; 6 x 9 in.; 161 pages. Available from the Superintendent of Documents, Washington, D. C. Price, 30 cents.*]

BLUEPRINTS. This book affords a basis for study of blue print reading which, together with practice in the actual reading of drawings, will guide the student to proficiency in this important subject. While the book has been compiled and published primarily for welders, it contains information of value for anyone concerned with mechanical construction. With the help of the book, the student can learn with a few hours spare-time study what it would take many months to pick up in the ordinary course of every-day work. It will enable welders, mechanics and others to answer questions for themselves, thus saving the time of the welding supervisor or foreman. [*"Simple Blueprint Reading with Special Reference to Welding." Published by the Lincoln Electric Company, Cleveland, Ohio. Semi-flexible simulated leather cover; 6 x 9 in.; 138 pages. Price, 50 cents in U.S.A. and 75 cents elsewhere.*]

ECONOMIC PIPE SIZE. The economic pipe size, for which the sum of pipe and pumping costs is a minimum, are derived for both the turbulent and viscous regions of flow. The resulting equations are represented by convenient Nomographs. By solving the optimum-diameter equations simultaneously with the critical Reynolds number, a convenient relation has been found to indicate whether any given flow will be turbulent or viscous in a pipe of optimum diameter. Although the optimum velocity of many liquids in turbulent flow runs from 3 to 4 feet per second, much lower optimum velocities are calculated for very viscous liquids. [*"Economic Pipe Size in the Transportation of Viscous and Nonviscous Fluids," by B. R. Sarchet and A. P. Colburn. Published in Industrial and Engineering Chemistry, Room 706, Mills Building, Washington, D. C., September, 1940, volume 32, No. 9, pages 1249 to 1252. Price, single copies, 80 cents.*]

RUSSIAN PRACTICE. A Russian textbook on heating and ventilating. Includes information on methods of heating such as furnace, stove, gas, and electric; pipe fittings, radiators, both forced and gravity circulation hot water heating; steam heating; and instruments. The chapter on hot water heating systems gives information on the steam jet method of circulation which is quite popular in the Soviet Union. [*"Heating and Ventilating," by B. M. Ashe. Published in Russian; 514 pages; 328 illustrations. For sale by the Four Continental Book Corporation, 255 Fifth Ave., New York, N. Y. Price, $2.30.*]

CIVIL ENGINEERING HANDBOOK. A second edition published primarily to bring the text into conformance with present practice in the civil engineering field. In some sections the material has been considerably rearranged while in others it has been almost entirely re-written. Of interest to heating and ventilating engineers are the sections on mechanics of materials, hydraulics, stresses in framed structures, concrete, foundations, and water supply. [*"Civil Engineering Handbook," by L. C. Urquhart, C.E. Second edition published by McGraw-Hill Book Company, Inc., 330 West 42nd Street, New York, N. Y. Semi-flexible binding; 6 x 9 in.; 877 pages. Price, $5*]

EDITORIALS

Fuel Estimating

It does not seem unreasonable to expect that with the increasing preparations for defense fuel prices may gradually rise. If this happens the corollary will be that greater attention will again be turned to the matter of insulating buildings in order to conserve fuel and reduce fuel costs.

It is being realized more and more that insulation does not only conserve fuel but that it increases the comfort of the occupants of the building. This point was emphasized in several papers presented at the centennial meeting of one of the insulation groups some months ago. Greater emphasis is being placed on this point now than ever before, probably because the public is already well sold on the fuel savings possible with insulation.

At the same time, the problem of estimating the heat loss of insulated and uninsulated buildings still leaves something to be desired from the standpoint of accuracy. At the meeting referred to above, an architect with Tennessee Valley Authority reported tests on electrically-heated houses where the savings possible from insulation as computed were about one-third higher than those that actually resulted. This reflects no discredit whatever on the value of insulation; as the author pointed out, the discrepancy was probably due to the fact that the heat loss, and consequently the fuel consumption, over a season is less than estimated by the usual methods because the house is not heated at night. While allowance can be made for this with reasonable accuracy in the case of the automatic fuels, the problem is a little more complex with hand-fired coal plants. The tendency for the engineer is to estimate the fuel consumptions of both the insulated and uninsulated house somewhat higher than they actually are with the result that somewhat higher dollar savings are computed than actually accrue.

It may be that the problem of estimating fuel consumption is still further complicated by the important factor of infiltration. The infiltration is originally computed at, say, a 15-mile wind velocity under design conditions and, generally speaking, the methods used for estimating fuel consumption carry this loss proportionally through the whole heating season. This is probably not the case. Actually the wind not only varies in velocity, sometimes being zero, but even reverses its direction, thus considerably changing the computed heat loss and resulting fuel consumption.

Two recent papers—one from the district heating and one from the gas heating fields—seem to indicate a revival of interest in the cubage of a building as a unit on which to base heat loss and fuel consumption. The tendency in past years has been to stress the dangers and inaccuracies of a cubage unit in this connection. However, any unit has its faults but the advantages of the cubage unit are so great that it will probably never be abandoned by engineers.

Probably the only reason why the present methods of estimating fuel consumption are within striking distance of the actual consumption is that the total heat loss is made up of so many different factors that the errors in both directions may compensate for each other.

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Reclaimed Houses

The housing shortage, immediately apparent in some cities adjacent to military and naval ports and to certain defense industrial plants, is beginning to make itself felt in almost all such cities. The potential business in heating in housing under way or projected to overcome this shortage is obvious; not so obvious but nonetheless real is the market for modernizing existing houses which in many cases have been idle for a considerable time.

The best available data show that the business in modernizing and maintenance of heating systems ordinarily at least equals that in new construction. Under present circumstances, with the possibilities of higher labor costs and heavy demand on materials, the modernizing of foreclosed and abandoned houses may become relatively more attractive than now to prospective buyers than new homes.

Thus there may be a large volume of modernizing work immediately ahead. Let us hope it will be profitable as well as voluminous.

◆

Poultry Housing

A prominent agricultural engineer has pointed out that poultry house designs are still made on cut and try methods; that there is fairly good agreement on use requirements, but that no authority has gone on record as to what inside temperature and humidity variations are tolerable. He urges a rational approach to this problem.

Since the primary purpose of a poultry house is to protect birds from extremes of temperature, humidity, and wind, the lack of logical design data and practice is hard to understand, especially since so much attention, relatively, has been given to the problems of dairy stable ventilation, heating, and even air conditioning.

It is encouraging that a joint committee on poultry housing has been set up by the American Society of Agricultural Engineers and the Poultry Science Association. Perhaps this committee will set up performance standards on poultry houses to clarify the present situation.

WATER REQUIREMENTS OF HOT WATER HEATING SYSTEMS

TABLE 1.—WATER CIRCULATION NECESSARY IN HOT WATER HEATING SYSTEMS FOR VARIOUS DROPS IN TEMPERATURE THROUGH THE HEATING SURFACES

Drop in Water Temp. through Radiator, F	Gallons per minute (GPM) of Water Necessary			
	Per Sq. Ft. of Radiator Rating			Per 1000 B.t.u. per hour of heat load, Mbh.*
	Usual Gravity Rating[1]	Intermediate Rating*	Steam Rating*	
	150 B.t.u. per hr.	200 B.t.u. per hr.	240 B.t.u. per hr.	1000 B.t.u. per hr.
10	0.0329	0.0417	0.0500	0.2083
11	0.0299	0.0379	0.0454	0.1896
12	0.0274	0.0347	0.0417	0.1736
13	0.0253	0.0320	0.0385	0.1600
14	0.0235	0.0298	0.0357	0.1488
15	0.0219	0.0277	0.0333	0.1388
16	0.0201	0.0260	0.0312	0.1300
17	0.0194	0.0245	0.0294	0.1225
18	0.0183	0.0231	0.0277	0.1157
19	0.0173	0.0220	0.0263	0.1100
20	0.0164	0.0208	0.0250	0.1040
21	0.0156	0.0198	0.0238	0.0992
22	0.0149	0.0189	0.0227	0.0947
23	0.0143	0.0181	0.0217	0.0906
24	0.0137	0.0174	0.0208	0.0868
25	0.0132	0.0167	0.0200	0.0833
26	0.0126	0.0160	0.0192	0.0800
27	0.0122	0.0154	0.0185	0.0772
28	0.0117	0.0148	0.0178	0.0744
29	0.0113	0.0144	0.0173	0.0718
30	0.0110	0.0139	0.0167	0.0694
31	0.0106	0.0134	0.0161	0.0672
32	0.0103	0.0130	0.0156	0.0650
33	0.0099	0.0126	0.0151	0.0631
34	0.0097	0.0123	0.0147	0.0613
35	0.0094	0.0119	0.0143	0.0595
36	0.0092	0.0116	0.0139	0.0580
Column No.	1	2	3	4

[1]Computed at 170F with water at 8.12 lb. per gal.
*Computed at 210F with water at 8.0 lb. per gal.

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WATER REQUIREMENTS OF HOT WATER HEATING SYSTEMS

The table on the other side of this page is derived from the essential relationship between the quantities of heat, the weight of water heated, the rise in temperature of the heated water, and the specific heat. The basic equation can be given as

$$W = \frac{H}{\text{Sp. Ht. } (t_2 - t_1)}$$

where

W = Quantity of water flowing in lb. per hr.
H = Quantity of heat in B.t.u. per hr.
t_2 and t_1 = Degrees F before and after heating, respectively
Sp. Ht. = Specific heat of water.

It is characteristic of hot water heating systems that they cannot give up heat to the air in the room unless sufficient quantities of water are passed through the radiators so that as the water cools a great enough quantity of heat is emitted to let the radiator maintain its temperature. Hot water heating systems are commonly designed for gravity flow with water entering the radiator at about 180F and leaving it at 160F. This gives a 20F drop in temperature through the radiator and the average temperature of the water is 170F. Where water is pumped through the radiators, the amount passing through depends on the pressure difference, and common practice permits considerably greater amounts of water to pass through a given radiator than can be passed with gravity flow. The result is that with forced circulation, the temperature drop through the radiator may be smaller than in the case of gravity radiators, so in order to keep down the size of radiators required, common practice is to raise the temperature of the circulating water to perhaps 200F and in some cases to as high as 210F or 215F if the system is put under pressure.

The table on the other side of the sheet has been prepared with columns indicating each of the temperature conditions commonly encountered with the drops in temperature usually found. The table is arranged on the basis of the unit heat requirements both in terms of gravity ratings, steam ratings, and per 1000 B.t.u. per hr.

The following is an example of the use of the table:

Example: How much water at an average temperature of 210F must be passed through a forced hot water radiator having a steam rating of 60 sq. ft. if the drop in temperature through the radiator is 20F?

SOLUTION: Enter Table 1 at 20 in the first column and move horizontally to column 3 where the figure 0.0250 will be found; multiply this by 60 and get 1.5 as the number of gallons of water which must be passed through the radiator each minute.

Examination of the table indicates that if the temperature drop through the radiator in the previous example had been only 10F it would have been necessary to pass 3 gallons per minute through the radiator in order to maintain the same heat output from the radiator.

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NEWS OF THE MONTH

Refrigeration Equipment Makers Prepare for Fall Meeting; Gott and MacDonald Scheduled to Speak

CHICAGO—Announcement of the list of speakers, their subjects and the schedule of sessions for the 1941 fall meeting of the Refrigeration Equipment Manufacturers Association at French Lick, October 10-11, has been made by J. D. Colyer, vice-president, Wolverine Tube Company, and chairman of the program committee.

On the program proper four nationally prominent speakers in their respective fields will present formal papers on topics now very much in the minds of executives in the industry. In addition to the subjects which speakers will develop, several of the association's committees will submit reports on other matters of immediate interest to everyone in the industry and trade. Many of these reports will be whipped into final form for presentation in committee sessions to be held at French Lick October 9—a day in advance of the general meetings opening on Thursday morning, October 10, the first of which will immediately follow a breakfast meeting of the board of directors that morning.

At 9:30 a.m., Mr. Colyer will call the first general session to order. Two formal addresses are scheduled for the forenoon meeting, the first on the subject of "Adjusting Association Activities to Current Industry Problems" by Philip P. Gott, manager of the trade association department, U. S. Chamber of Commerce, Washington.

Following the discussion of Mr. Gott's paper, the second subject, "The Wholesaler—Our Partner in Business" will be presented by N. J. MacDonald, Elizabeth, New Jersey — an authority in the field of industrial merchandising. Mr. MacDonald is vice-president of Thomas & Betts Co., manufacturers of electrical supplies. The remainder of the session, which will run until 1 p.m., will be devoted to action on committee reports. The afternoon is left open for recreation and relaxation.

The Friday forenoon program will be opened by an address on "Requisites for Success in a Jobbing Business" by J. W. Baillie, secretary-treasurer, Detroit Lubricator Company, and chairman of the association's credit managers' division. The second paper of the session on the subject, "Cold Facts and Hot Ideas—What Advertising Can do for You," will be given by Stuart G. Phillips, advertising manager, Dole Valve Company. Mr. Phillips, long active in the activities of the National Industrial Advertisers Association, will bring a message based on some twenty years experience in the industrial advertising field.

Following the discussion of his paper reports from the officers and remaining committees will be considered.

Among the many matters of importance to be taken up will be final arrangements for the Third Annual All-Industry Refrigeration and Air Conditioning Exhibition, sponsored by REMA, to be held at the Stevens Hotel, Chicago, January 13-16, 1941. Exhibition manager R. M. McClure reports that already nearly as many exhibitor firms have signed up for a space in the 1941 All-Industry Exhibition as took part in the 1940 show last January. It is expected that the 1941 show will break all previous records.

From an attendance standpoint the 1941 show is expected to exceed by several thousand the number of visitors who turned out last January. All of the cooperating technical societies and national trade associations who held their 1940 conventions and meetings in Chicago during the week of the All-Industry Exhibition are planning big meetings again in Chicago for the week of January 13-16 next to give their members an opportunity of also taking in the show. Practically all of these meetings will be held in the Stevens Hotel, in which the REMA show again will be staged.

One of the important questions on the docket for action at the fall meeting has to do with the place and date for holding the Fourth All-Industry Refrigeration and Air-Conditioning Exhibition. Instead of waiting until next January to discuss the matter of place and date for the succeeding year's show, at which time the decision would probably have to be made by the directors as in the past, it is thought best to throw this matter open for discussion and decision by the membership.

There is sentiment in favor of rotating the All-Industry Show from year to year or every other year; and perhaps equally as strong a sentiment for holding it permanently in one central location. One of the most important aspects of the question is whether or not the show should be held in a location where both the exhibition and the conventions which meet concurrently with it shall be housed under the same roof.

Preceding the formal sessions of the fall meeting, members of the following committees will meet Wednesday at the French Lick Springs Hotel:

Jobbers Relations Committee: Chairman, J. S. Forbes, Superior Valve & Fittings Co.; F. J. Hood, Ansul Chemical Company; and K. B. Thorndike, Detroit Lubricator Company.

Membership Committee: Chairman, M. R. Oberholzer, L. H. Gilmer Company; Ivan Corcoran, Square D Company; H. A. Chandler, McIntire Connector Co.; T. C. Davis, The Dayton Rubber Mfg. Co.; and R. O. White, Day & Night Manufacturing Company.

RSES Relations Committee: Chairman, John Wyllie, Jr., Temprite Products Company; K. M. Newcum, Superior Valve & Fittings Co.; and H. W. McPherson, The Electrimatic Corp.

Trade Practices Committee: Chairman, W. C. Allen, Modern Equipment Corp.; L. F. Blough, White-Rodgers Electric Co.; R. W. Kritzer, Peerless of America, Inc.; Robert LeBaron, Virginia Smelting Co.; and J. A. Strachan, Kerotest Mfg. Co.

Statistics & Standards Committee: Chairman, Mr. Corcoran; C. M. Brown, Tecumseh Products Company; W. D. Keefe, Fedders Mfg. Co.; Mr. LeBaron; A. B. Schellenberg, Alco Valve Company; and A. B. Newton, Minneapolis-Honeywell Regulator Company.

Exhibition Committee: Chairman, M. W. Knight, Peerless of America, Inc.; C. H. Benson, Imperial Brass Mfg. Co.; D. H. Daskal, Perfection Refrigeration Parts Co.; R. H. Luscombe, Penn Electric Switch Co.; and Mr. Thorndike.

Registration Committee: Chairman, Barrett Scudder, Jas. P. Marsh Corp.; J. W. Hatch, Bush Manufacturing Company; and A. J. Male, Bonney Forge and Tool Works.

ASRE Relations Committee: Chairman, Mr. Schellenberg; E. W. McGovern, E. I. duPont de Nemours; and F. L. Riggin, Jr., Mueller Brass Co.

Ordered to Stop Use of A.C. Term

WASHINGTON — The Federal Trade Commission has ordered that Air Conditioning Textiles, Inc., a corporation, cease and desist from using the term Air Conditioning the Human Body or any other word to designate or describe its soap or otherwise representing that said soap possesses air conditioning properties; representing that said soap reduces body temperature or reduces humidity or eliminates perspiration objections; or representing that the human body breathes through the pores of the skin.

Stoker Exhibit for Chicago Mart

CHICAGO—L. G. Briggs, president of the Midwest Stoker Association, reports that arrangements have been completed with the world-famous Merchandise Mart for a permanent stoker and automatic coal heat display on the Home Building Exhibit floor of the Mart.

Massachusetts Education Department Opens H & V Courses; Seven Subjects to be Taught by Specialists

BOSTON—C. P. Yaglou, associate professor of Industrial Hygiene, Harvard School of Public Health, will conduct a class on Thursday Evenings, beginning Oct. 17, in "Thermodynamics of Air Conditioning," a series of sixteen lectures in two parts. This course, is one of a number sponsored by the Massachusetts Department of Education, deals with the application of simple thermodynamics to the solution of practical air conditioning problems, and is intended for consulting engineers and designers. Among the subjects to be discussed are: physical and chemical changes in the air of occupied rooms; heat and moisture exchange between the human body and the environment; comfortable and healthful air conditions; with heat and moisture transfer in various types of air conditioning systems, together with designs for complete air conditioning systems for a theatre and for a textile mill or a mine.

P. A. L. Foulds will give again this fall, beginning Oct. 24, a course of eight lectures on "Practical Air Conditioning for Residences, Restaurants, Stores and Officers," covering the fundamentals, both theory and practice.

On Oct. 29 C. Wesley Nelson, instructor in mechanical engineering at Wentworth Institute, begins a series of fifteen lectures Tuesday evenings in air conditioning for factories, theatres and other buildings.

Joseph Greenblatt, consulting engineer will conduct a class in heating and ventilating of sixteen lectures, for men in the trade having to do with installation, design, maintenance and selling of heating and ventilating apparatus and systems. It also includes air washers and air conditioning and is also offered by correspondence. Classes are held Monday evenings, starting Oct. 28.

John H. Barrett, sales engineer, Minneapolis-Honeywell Regulator Company, will have a class of eight lessons on Monday evenings, beginning Nov. 18, in automatic controls for heating, cooling and air conditioning, covering coal, oil and gas burners and air conditioning systems, illustrated by films showing individual controls, assembly of systems and views of installations.

Alfred Herschel, research engineer and supervisor of training, The Foxboro Company, will conduct a class Tuesday evenings, beginning Oct. 22, of sixteen lessons on instrument craftsmanship. This course is designed to instruct the student in the various industrial instruments in common use and is intended to be of practical value to those who have the responsibility of servicing and maintaining these instruments. The lectures will be non-technical and will cover constructional details and calibration methods. Classroom models and demonstrations will be used for illustration. Mr. Herschel will also conduct an advanced course of instrument craftsmanship for eight lectures on Thursday evenings. It is offered to those who have equivalent knowledge covered by the preceding course, and begins Oct. 24.

Endorse USHA Defense Program

WASHINGTON—Mayors of 180 American cities having a total population of more than 43,000,000 have gone on record as recommending that the United States Housing Authority and local housing agencies carry on the major part of the national defense housing program and urge that adequate funds be made available without delay, according to Nathan Straus, USHA Administrator, following a visit from Mayor Brantley Harris of Galveston, Texas. Mr. Harris is a member of the United States Conference of Mayors, who tendered Mr. Straus a resolution adopted by the mayors in New York City endorsing the low-rent and slum clearance program conducted by local housing authorities with the aid of the USHA.

The resolution indicates that the mayors feel that USHA has stepped most quickly into the service of defense housing needs, commenced about 20 representative defense housing projects throughout the country, and pursued these projects so actively that most of them will be ready for occupancy in the first months of 1941 or earlier.

Gas Heating Men to Meet October 7

ATLANTIC CITY—The Association of Gas Appliance and Equipment Manufacturers will hold its annual meeting October 7 at the Hotel Claridge here. The meeting will be held simultaneously with the convention of the American Gas Association.

The principal event of the meeting of the Association of Gas Appliance and Equipment Manufacturers will be the election of officers and directors for the next fiscal year. Lyle C. Harvey, president of the Bryant Heater Co., Cleveland, Ohio, is chairman of the nominating committee. Frank H. Adams, president of the Association and vice-president of the Surface Combustion Corp., Toledo, will give the address of welcome.

The Association's Gas House Heating and Air Conditioning Equipment Division and its sub-groups will hold meetings on Sunday, Oct. 6, prior to the actual opening general session. W. L. Seelbach, secretary and treasurer of the Forest City Foundries Co., Cleveland, is chairman of this division. Among the sub-groups of this committee are: Gas Boiler Group, R. L. Blodgett, secretary-treasurer, Pennsylvania Furnace & Iron Co.; Gas Furnace Group, Harold Massey, American Radiator and Standard Sanitary Corp.; Gas Conversion Burner Group, J. C. Bowery, general sales manager, Roberts-Gordon Appliance Corp., Buffalo.

On Tuesday, Oct. 8, the Gas Floor Furnace Group of the Association's Gas House Heating and Air Conditioning Equipment Division will hold a breakfast meeting with Clarence Coleman, president of the Coleman Lamp & Stove Co., Wichita, Kansas, presiding.

Fan Test Demonstration. A standard fan test set up in the showrooms of the Commonwealth Edison Company, Chicago, as part of an air conditioning and ventilating show held annually by that utility. The demonstration was put on by Ilg Electric Ventilating Company which had two of its engineers supervising the demonstration.

Pacific Coast to Have H&V Show

SAN FRANCISCO—The Pacific Heating & Air Conditioning Exposition will be held in the Exposition Auditorium in the Civic Center here June 16-20, 1941. Coincidentally, the summer meetings of the ASHVE and the Heating, Piping & Air Conditioning Contractors National Association will be held here.

This Exposition will be a special interim showing between the normal sequence of the 6th and 7th International Heating & Ventilating Expositions, the 6th having been held at Cleveland in January of this year and the 7th will be held at the Commercial Museum and the Convention Hall in Philadelphia, January 26-30, 1942. It will not disturb the biennial sequence of those expositions.

The management of the exposition has announced that of the 124 spaces in the first area which has been offered, 30 have already been reserved. International Exposition Company will conduct the Exposition as they have previous International Heating & Ventilating Expositions. Permanent headquarters are in Grand Central Palace, New York, where communications should be addressed.

USHA Speeds Defense House Plans

WASHINGTON — Cutting in half the time normally required to prepare architectural plans for a defense housing project, the United States Housing Authority has completed working drawings and standard specifications quickly adaptable to the requirements for various localities throughout the nation.

This material, prepared by the USHA technical division, is designed to assist local architects on defense housing projects. Its use will enable them to conserve as much as 50 per cent or more of the time normally required to complete a set of working drawings. It includes a series of superstructure working drawings to which the local architect need add only the foundation requirements and prepare the necessary site and utility plans; a standard specification dummy; and directions for using plans and specifications.

The first of the series of superstructure working drawings, which is now available to local housing authorities where defense housing projects are located, is for homes built of frame construction. With the exception of flats, all are designed for individual heating systems using coal fuel.

The USHA Technical Division also has under way plans for other types of dwelling units, different types of construction, and varying heating systems and fuel. These will give a wider choice to meet specific local housing project requirements. Detail grouping of units for site layouts and possibilities for the development of the site plan will be shown.

Gas Heat Institute Formed on Coast

LOS ANGELES—With a membership composed of local manufacturers and distributors of heating equipment made in other sections of the country, the Institute of Gas Heating Industries has been organized here, with a membership, as of September 13, of 20.

First permanent officers are J. S. Andrews, Andrews Heater Co., President; J. Benjamin, Ashburn Heating Co., vice-president; and Harvey Aikins, Mayfair Furnace Co., secretary-treasurer.

The organization has no board of directors. The entire membership is divided into four divisions — space heater, forced air and gravity furnaces, floor furnaces, and direct-heating. Each of these divisions is composed of members engaged in a particular phase of heater manufacturing or distribution.

The initial objectives of the new organization are:

(a) Formulation and preparation of a warm air furnace heating ordinance;

(b) Instrumentation of a program whereby heater manufacturers would be given a voice in determining the rules of A.G.A. heater tests; and

(c) Development of a program for standards of guarantee, collective handling of labor, exchange of credit information, working out of a uniform credit setup for each member, and the standardization of installations.

Committees already appointed and active are the Gas Ordinance Committee, Ray Logue, chairman, J. C. Redder and J. Stanley Johnston; Standardization and Coordination of Installations Committee, Clifford Menig, chairman, Fred Wambsganst and Glenn A. Ashburn; A.G.A. Committee, C. H. Pill, chairman, E. L. Payne and J. P. Esseck; Public Relations, Publicity and Entertainment Committee, J. Benjamin, chairman, G. C. McNulty and Thomas A. Martin; Guarantee Committee, Ray Logue, chairman, J. Benjamin and E. L. Payne.

Carrier to Address Gas Convention

ATLANTIC CITY — The twenty-second annual convention of the American Gas Association will be held here Oct. 7-10 at the Atlantic City Auditorium. Among the papers to be presented and the addresses to be made will be a talk by Dr. Willis H. Carrier, chairman of the Board, on Air Conditioning —An Industry Opportunity.

R. M. Conner, director of the A.G.A. Testing Laboratories in Cleveland, will present a paper on Design of Atmospheric Gas Burners.

Avery Engineering Co. Expands

CLEVELAND—Avery Engineering Company, Cleveland, has leased space at 1906 Euclid Ave. to enlarge its facilities to handle air conditioning and commercial refrigeration.

Coal Groups Endorse Research Plan

WASHINGTON — Appalachian Coals, Inc., the country's largest and pioneer coal marketing agency, and the National Coal Association have endorsed the broad cooperative program of scientific research of Bituminous Coal Research, Inc. Actual research work is scheduled to begin shortly since campaign subscriptions already total $121,500.

The National Coal Association has made a liberal subscription toward support of the research program and the board of directors of Appalachian Coals has urged member companies to subscribe through their local operators' associations. Both groups recommended that the coal industry as a whole, as well as affiliated interests, support the program.

To gain new and regain old markets for the coal industry is the aim of the program, funds for which are being pledged by coal operators throughout the country. Howard N. Eavenson, well-known coal operator and engineer of Pittsburgh, Pa., is president of the research corporation.

Embodied in the eight point research program are the problems of smokeless hand-firing of coal, completely automatic residential heating, dustproofing of coal, railroad locomotive fuels, publication of engineering bulletins, a standard smokiness index, modification of ash fusion characteristics, and emission of solids from stokers. The development of coal-consuming devices to improve the use of coal and to serve new markets will be the principal activity of the program. Manufacturers are being asked to cooperate in designing and manufacturing the new burners.

According to E. R. Kaiser, assistant to the president, Bituminous Coal Research, the research agency will also coordinate the research on coal throughout the country and keep executives and engineers of the coal industry constantly informed on trends in coal utilization and research by publication of periodical digests and engineering bulletins.

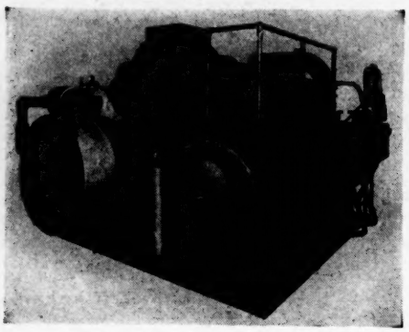
Edwin D. P. Greiner

NEW YORK—Edwin D. P. Greiner, district manager of the New York and New England distributor organization of Carrier Corporation, died suddenly September 10 of a cerebral hemorrhage in the St. Lawrence Hospital, Bronxville. He was 36.

Philip M. Patterson

WEST ROXBURY, MASS. — Philip M. Patterson, New England representative of The Torrington Manufacturing Company, died at his home here September 2.

NEW EQUIPMENT



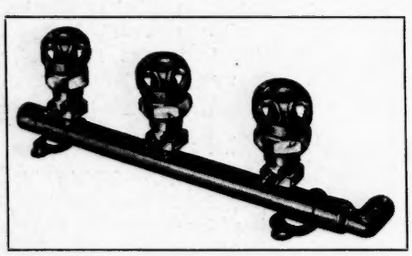
Refrigerating capacities from 150 to 650 tons.

Worthington Centrifugal Refrigeration Units

NAME—Centrifugal type refrigeration units.
PURPOSE—For air conditioning service.
FEATURES—This company is now offering a complete line of centrifugal refrigeration units for air conditioning service with capacities ranging from 150 to 650 tons. Speeds are said to be relatively low ranging from 4000 to 5400 r.p.m. and pressures are said to not exceed 15 lb. as a result of using Freon 11 as a refrigerant. A feature of these compressors is an auxiliary pump operating independent of the main drive for furnishing full pressure lubrication to all bearings during starting and stopping periods. In starting, the main power can be applied only after the auxiliary pump develops full pressure, and in stopping, the pump is automatically cut in as soon as the main power is shut off.
CAPACITIES—150 to 650 tons.
MADE BY—*Worthington Pump and Machinery Corporation, Harrison, N. J.*

Henry Packless-Valve Manifold

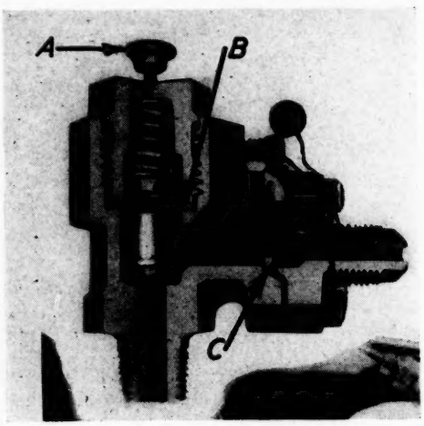
NAME — Henry packless-valve manifold.
PURPOSE—For the distribution and control of the flow of refrigerant.
FEATURES — Manifold serves threefold purposes: (1) a compact and convenient means of mounting shut-off valves, (2) more even distribution of refrigerant through valves and lines and, (3) centralized manual control location. These manifolds feature the use of Henry balanced action diaphragm packless valves which are said to be positive opening under all pressures and conditions. The angle valves are mounted on the manifold with extruded reinforced silver soldered joints. Manifolds are made with from 2 to 6 valves with either flare or solder connections.
SIZES—From ¼ to ⅝ in. SAE flare, or OD solder. Sizes of elbow manifold connection range from ⅜ in. SAE flare to 1⅜ in. OD solder.
LITERATURE AVAILABLE—Bulletin 129.
MADE BY — *Henry Valve Company, 1019 N. Spaulding Ave., Chicago, Ill.*



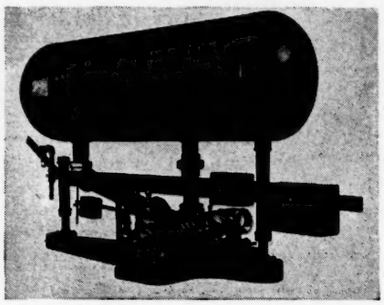
Henry manifolds.

Black-Sivalls & Bryson Refrigerant Relief Valve

NAME—Safety head relief valve.
PURPOSE—For preventing dangerous over-pressures in refrigerating systems.
FEATURES — After the unit is installed on a pressure system, a slight lift on button (A) opens a relief valve which is then held open by gas pressure on the actuating diaphragm (B) until rupture disc (C) is burst by emergency over-pressure. Bursting of the rupture member by over-pressure removes the pressure from the actuating diaphragm (B) and the valve becomes an ordinary spring pop valve, closing when the emergency is past and pressure has dropped to normal. The spring relief valve then holds the gas in the system with a minimum of leakage, and a new rupture disc may be installed at the operator's convenience without shutting down the equipment.
MADE BY—*Black-Sivalls & Bryson, Inc., Kansas City, Mo.*



For preventing over pressures on refrigeration.



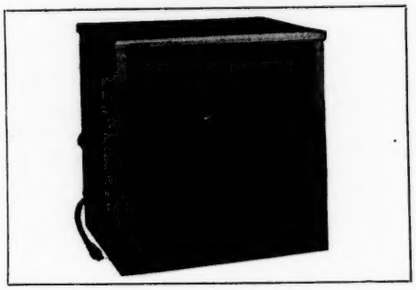
For steam drainage and boiler feeding.

Acker Steam Drainage and Boiler Feeding System

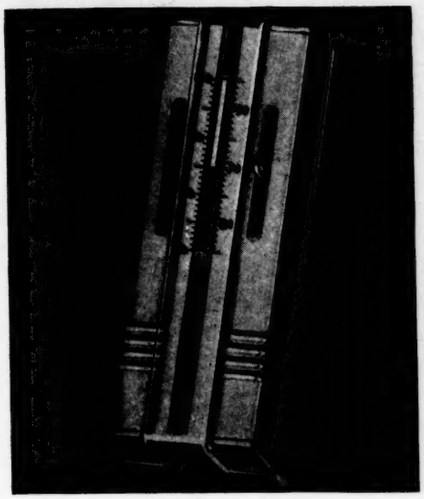
NAME—Acker Deaerating and Equalizing system.
PURPOSE—For feeding make up water and condensate to boilers.
FEATURES—These systems are said to automatically feed condensate or make-up water to boilers and at the same time deaerate all water fed to boilers. The systems are said to be adaptable to any conditions in plants operating up to 250 lb. pressure and are said to function without pumps or other mechanisms. Manufacturer states that they can be used with or without individual traps or with needle type valves. Condensate and make-up water can be handled at 212F and higher.
MADE BY—*W. M. Acker Organization, Inc., 3167 Fulton Road, Cleveland, Ohio.*

Alter Humidifiers

NAME—Humidome humidifiers.
PURPOSE—For adding moisture to the air.
FEATURES—1941 line of Humidome humidifiers consists of three models, with humidifying capacities ranging from 3 to over 4 pints per hour. All of these units utilize a "Miracle Drum" which slowly revolves through water in the reservoir and is said to expose over 30 sq. ft. of evaporative surface to the air stream. Units have a reserve capacity of 5 gal. Manufacturer also states that these units will remove a considerable amount of dust and dirt from the air.
MADE BY—*The Harry Alter Company, 1728 So. Michigan Ave., Chicago, Ill.*



Model 341 Humidome.



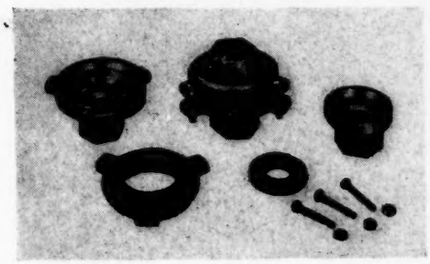
Made of transparent plastic.

Gleason-Avery Thermostat

NAME—Gleason-Avery thermostat.
PURPOSE—For controlling room temperatures.
FEATURES — Thermostat is designed for greater eye appeal and quicker and easier readability. It is said to be the first application of transparent plastic in the thermostat field. A single temperature scale is provided for both reading and setting with a finger tip adjustment. The lever is adjustable to the thermometer scale over 40F in temperature and is said to be sensitive to limits of 1F.
MADE BY — *Gleason-Avery, Inc., Auburn, N. Y.*

Patterson-Ballagh Swing Joint

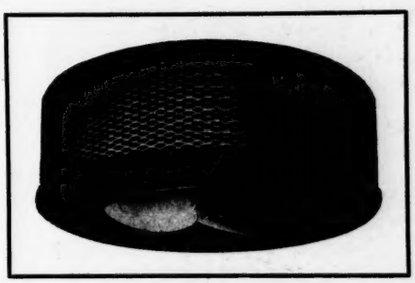
NAME—Patterson-Ballagh swing joint.
PURPOSE—For providing a flexible connection in steam lines.
FEATURES—Manufacturer states that this joint may be used upon lines with steam pressures up to 100 lb. Joint is said to be made of heavy bronze and to have a sealing unit which is a specially shaped and compounded rubber gasket. This gasket contains a percentage of metallic lead milled directly into the rubber. It is claimed that the gasket expands slightly under the heat which helps to maintain a perfect seal.
SIZES—¾ in. straight style.
MADE BY—*Patterson-Ballagh Corporation, 1900 East 65th Street, Los Angeles, Cal.*



Swing joint for steam lines.

McCord Vertical Projection Unit Heater

NAME — McCord vertical projection unit heater.
PURPOSE—For supplying heated air vertically downward.
FEATURES—Manufacturer states that the circular shape vertical type heater is particularly adapted to the McCord spiral fin tube construction because the shape and design of the fin creates air turbulence without undue restrictions, thus increasing heat transfer. The copper fins are metallically bonded to the copper tubes. The heating element has been tested to 2000 lb. hydrostatic pressure and is guaranteed for 200 lb. saturated steam pressure. The entire unit has been designed around Torrington quiet operating fan blades and the manufacturer claims that as a result the unit can be used in many installations not previously considered adaptable to vertical type heaters.
MADE BY—*McCord Radiator & Mfg. Co., Detroit, Mich.*



McCord vertical type unit heater.

Ray Heavy Oil Burner

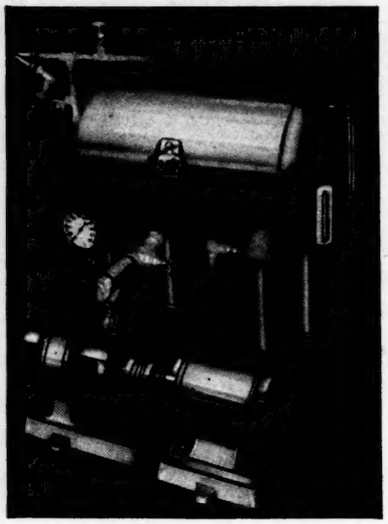
NAME—Ray automatic engineer.
PURPOSE—For burning No. 5 fuel oil in residential and small industrial applications.
FEATURES—A Liquid Bath Heater and Retractor oil control are two new developments incorporated in this burner to attain fully automatic operation of a pressure atomizing oil burner with No. 5 oil. The Liquid Bath Heater is said to operate on the indirect principle of the common kitchen double boiler in order to maintain a uniform oil temperature without cracking or carbonization. At the same time, the fluid bath acts to cool and protect the nozzle from the intense reflected heat of the firebox where carbonization of the oil may clog the nozzle. The Retractor oil control is designed to fill two primary functions —to act as the oil shut-off valve, and also to act as a pump to withdraw the oil from the nozzle and the nozzle fuel tubes after each shut-down of the burner, thus cleaning the fuel tubes and the nozzles after each firing period.
CAPACITIES—From 1½ to 6 gal. of oil per hr.
MADE BY—*Ray Oil Burner Co., 409 Bernal Avenue, San Francisco, Cal.*

Small oil burner for No. 5 fuel oil.

Friez CO_2 Indicator

NAME—Friez CO_2 indicator, Model Y/2.
PURPOSE — For determining the amount of carbon dioxide in flue gases.
FEATURES—Instrument is machined from a solid block of clear plastic, while the metal parts are of stainless steel. The absorbent solution is brilliant red in color and its level is reflected from the polished scale to allow the operator to line up the liquid level, its reflection and the scale graduations thus preventing any parallax. Instrument is said to be accurate to within ½ of 1%. Readings are taken by zeroing the scale, pumping a sample of gas into the chamber, and raising and lowering the absorption basket. Indicator is supplied complete with a carrying case in which space is provided for stack thermometer and full size draft gauge.
MADE BY—*Julien P. Friez & Sons, Division of Bendix Aviation Corporation, Baltimore, Md.*

For indicating carbon dioxide in flue gases.



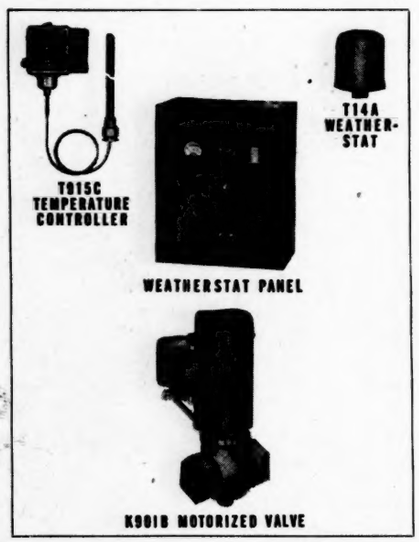
Westco boiler feed system.

Westco Boiler Feed Systems

NAME—Westco automatic boiler return systems.
PURPOSE—For returning condensate and supplying make-up water to boiler.
FEATURES — These systems are planned specially for boilers up to 750 hp. and 200 lb. pressure. They are designed to automatically return all condensate to boiler from both high and low pressure equipment, to maintain a uniform water level, to keep return lines constantly open, and eliminate back pressure on return lines. Each combination is equipped with a turbine-type pump having a patented renewable liner. Also incorporated is an automatic boiler water level control, a low water cut-off and alarm.
MADE BY—*Micro-Westco, Inc., Bettendorf, Iowa.*

M-H Zone Control System

NAME—Minneapolis - Honeywell Modulating Weatherstat zone control system.



Controls two-pipe steam heating systems.

PURPOSE—For controlling the operation of two-pipe steam heating systems.
FEATURES—The system employs the T14A Weatherstat as its master controller. This device is mounted on the outside of the building on the outer wall of the zone it is to control and functions like a miniature building in that it has its own thermostat and heating plant, the latter in the form of an electric heating coil. Manufacturer states that one of the outstanding features of this system is that it provides for the continuous flow of steam during lower outdoor temperatures and for two-position operation of the modulating motorized valve during higher outdoor temperatures. The fact that the zone valve is operated in a two-position manner during outdoor temperatures that require very little steam is said to eliminate distribution difficulties encountered by most modulating systems in mild outside temperatures. This system is applicable to two-pipe steam heating systems only and is recommended for installations where a constant source of steam is available and where a motorized steam valve can be installed. All radiators must be properly orificed and a relatively constant pressure differential must be maintained between the supply side of the motorized valve and the system return.
MADE BY — *Minneapolis - Honeywell Regulator Co., 2950 Fourth Ave., South, Minneapolis, Minn.*

Degree-Day Gauge

NAME—Weatherwise Dodson degree day gauge.
PURPOSE—To indicate degree days.
FEATURES—Manufacturer states that this device eliminates the use of readings at distant weather bureau points and makes it possible to obtain the daily degree days merely by reading the gauge once a day. Gauge is sold for use in both 65 and 70F base degree day operation.
MADE BY—*Degree Day Systems, 49 West 42nd Street, New York, N. Y.*

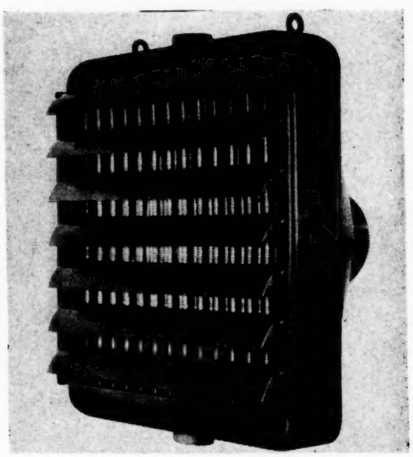
Marsh Automatic Vent Valve

NAME—Airovent No. 10 A, No. 20 A.
PURPOSE — For venting air from steam heating systems.
FEATURES—These valves are said to offer, at a popular price, many of the features which are incorporated in the higher priced Marsh line of valves. Some of the features include Marsh multiport silent venting, leak-proof and spit-proof operation, rapid venting of air and modern streamlined appearance. Airovent is made in two styles, No. 10 A automatic air valve and 20 A, air and vacuum valve. The vacuum feature includes a check which prevents return of air into the valve when condensing steam begins to form a vacuum. As the vacuum force increases the internal diaphragm expands and closes the valve.
MADE BY—*Jas. P. Marsh Corporation, 2073 Southport Ave., Chicago, Ill.*

Popular price vent valve.

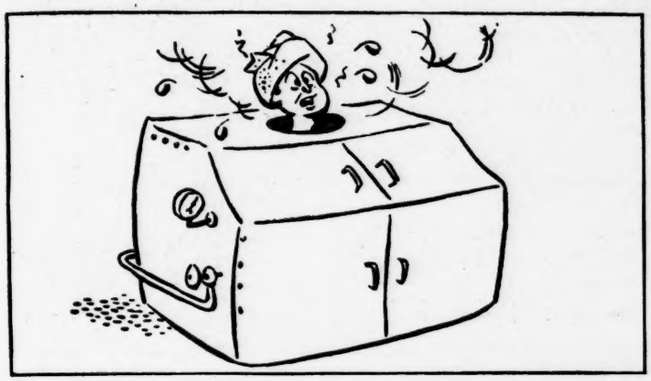
IIg Stream-Styled Unit Heater

NAME—IIg Stream-styled unit heater.
PURPOSE—For warming air.
FEATURES—Unit heater features a two-piece graduated header to assure trouble free operation on all steam pressures and to give balanced steam distribution. Entire core of the heater is made of copper and fins are flanged and pressed on tubes for permanent union. The bottom header floats on slotted coil retainer and spring washers, to permit expansion and contraction of the coil independent of the casing. Motor is of a self-cooled type, specifically designed for these heaters.
MADE BY—*IIg Electric Ventilating Company, 2850 North Crawford Avenue, Chicago, Ill.*



IIg stream-styled unit heater.

Brother, this picture-page can save you money!

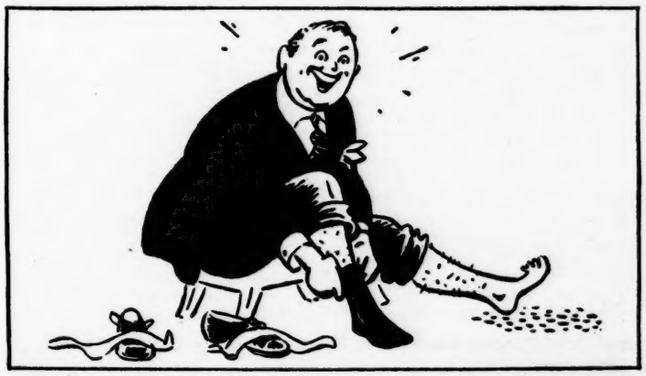


1 The treasurer can never put you in a sweat because of high filter costs. Dust-Stops* cost approximately 1-cent per CFM to install, including frames; approximately 1/10th-cent per CFM when filters need changing. A big saving over many other types of installations.

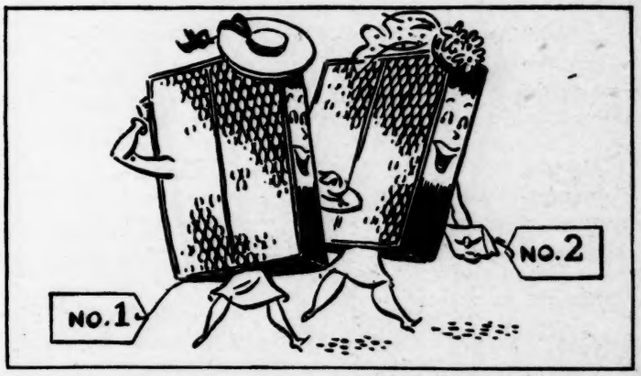
2 Let the plant firemen gather cobwebs, as far as Dust-Stops are concerned. Remember this *always*—Dust-Stops are safer, because the all-glass filtering medium, Fiberglas* fibers, and the patented adhesive will not support combustion.

3 Like the Northwest Mounties, Dust-Stops get what they go after. They arrest virtually all "nuisance" dusts. Capacity: 2 CFM per square inch of filter area at 300 FPM. Resistance (in inches of water gauge per inch of depth): .045 to .050 clean, and .11 to .12 dirty.

4 And the president can't get you out on a limb because the installation bleeds. The dust-catching adhesive on these filters will not deposit tiny droplets in the ducts or on the walls. And the fire-hazard from oil in the duct system is nil.



5 How much skilled labor is required to change socks? That's how easy it is to change Dust-Stops. This replaceable filter eliminates cleaning, charging, draining, expert supervision, and the need to keep a supply of spares. Dust-Stops fit all types of air handling systems.



6 You can pick up Dust-Stops practically anywhere. Your air conditioning manufacturer, jobber or dealer always has a supply on hand. No waiting! Order your supply today. Or write: Owens-Corning Fiberglas Corporation, Toledo, Ohio. In Canada: Fiberglas Canada, Limited, Oshawa, Ontario.

FIBERGLAS* DUSTOP* AIR FILTERS

*T. M. REG. U. S. PAT. OFF.

Made by Owens-Corning Fiberglas Corporation, Toledo, Ohio

SEE FIBERGLAS MADE: GLASS CENTER, NEW YORK WORLD'S FAIR, 1940

WITH THE MANUFACTURERS

The C. A. Olsen Manufacturng Company, Elyria, Ohio, has appointed *Edward P. Hayes* as vice-president. For many years Messrs. Olsen and Hayes were both associated with the Fox Furnace Division of American Radiator & Standard Sanitary Corporation, of which Mr. Olsen was president and Mr. Hayes was in charge of sales promotion and sales. The announcement indicates a resumption of this association after an interlude during which Mr. Hayes served as vice-president of May Oil Burner Corporation, Baltimore.

The Celotex Corporation, Chicago, has elected *L. R. Boulware* as vice-president and general manager. Mr. Boulware was formerly vice-president and general manager of Carrier Corporation.

Cochrane Corp., Philadelphia, Pa., has appointed *William C. Bennett* manager of its newly created appliance sales division. *John E. Fearon* has been placed in charge of the flow meter sales section to coordinate sales and engineering, initiate a program of new product development, and assist Mr. Bennett both in maintaining close personal contact with sales representatives and in the developing of meter sales promotion. *Edward J. Glinske* is placed in charge of application engineering, proposals, and order details in the specialties section.

Domestic Engine & Pump Company, Shippensburg, Pa., has appointed *Berchem Company,* 50 Church St., New York, to represent it in the sale of Ames vacuum heating pumps and condensate return pumps in New York and vicinity.

General Controls Co., Glendale, Calif., has moved into a new and permanent factory at 801 Allen Ave., Glendale. The new plant houses the general offices, research and testing laboratories, and manufacturing facilities.

Ilg Electric Ventilating Co., Chicago, Ill., has appointed *P. D. Briggs* as general sales manager. He will continue to reside in New York and will continue as vice-president and eastern manager. Mr. Briggs joined Ilg in 1917 as sales engineer. In 1928 he was appointed vice-president and eastern manager.

Janette Manufacturing Company, Chicago, Ill., has appointed *Stearns, Perry* and *Smith,* 51 Chardon St., Boston, as district agent for that territory.

Liberty Foundry Co., St. Louis, manufacturers of warm air furnaces, has completed a remodeling job at its plant at 2800 Ohio St., at a cost of approximately $20,000. The building contains 85,000 square feet of floor space and, in addition to manufacturing facilities, houses the general office and sales department.

May Oil Burner Corporation, Baltimore, has appointed *Charles R. Collins* as sales manager. He will be in charge of sales for all Quiet May oil heating equipment. He replaces Edward P. Hayes who has resigned.

Plandaire, Inc., Pittsburgh, Pa., has appointed *Air Conditioning Utilities Co.,* 8 West 40th St., N. Y., as its distributors in metropolitan New York. The company has also appointed *Glegge Thomas,* 1426 G St., N.W., Washington, D. C., as its representative in the Washington territory. *Hucker-Pryibil Company,* 1700 Walnut St., Philadelphia, has been appointed representative in that area.

Research Product Corp., Madison, Wis., has appointed *H. M. Sawrie, Jr.,* P. O. Box 147, Chattanooga, Tenn., as its sales representative in Tennessee. *Frank C. Fassett,* P. O. Box 9298, Sulphur Springs Station, Tampa, Fla., is the company's new sales representative in Florida.

Sterling Pump Corp., Hamilton, Ohio, has acquired the turbine pump division of the Roots-Connersville Blower Corp. of Connersville, Ind. A transfer of inventory has already been made from Connersville to Hamilton, and all former Roots-Connersville installations will now be serviced, if and when necessary, by an enlarged staff of Sterling engineers nationally available.

Henry J. McKenzie

Sterling Pump Corp., Hamilton, Ohio, has appointed *Henry J. McKenzie,* president. Mr. McKenzie has been executive vice-president and general manager. He will succeed *Maurice Rothschild,* who will continue as a director. Mr. McKenzie, who also retains his duties of General Manager in addition to those of president, has long been identified with and is an outstanding and well-known personality in the pump industry. Starting in 1917 in the engineering department of Layne & Bowler Corp., Los Angeles, Mr. McKenzie rose successively through the positions of district manager, general manager, and sales manager at Memphis, then was appointed general manager and vice-president of that organization. Later, he became manager of the turbine pump division of Worthington Pump & Machinery Corp., at the Holyoke plant.

Wolverine Tube Company, Detroit, has appointed *A. S. Kingerley* to assist *J. D. Colyer,* sales manager, in all jobber sales activity of the company. Mr. Colyer has served Wolverine in various capacities for 14 years, during the last three years of which he has been engaged in tubing sales in the Detroit area. In this work he will be succeeded by *H. L. Millett* of the sales department.

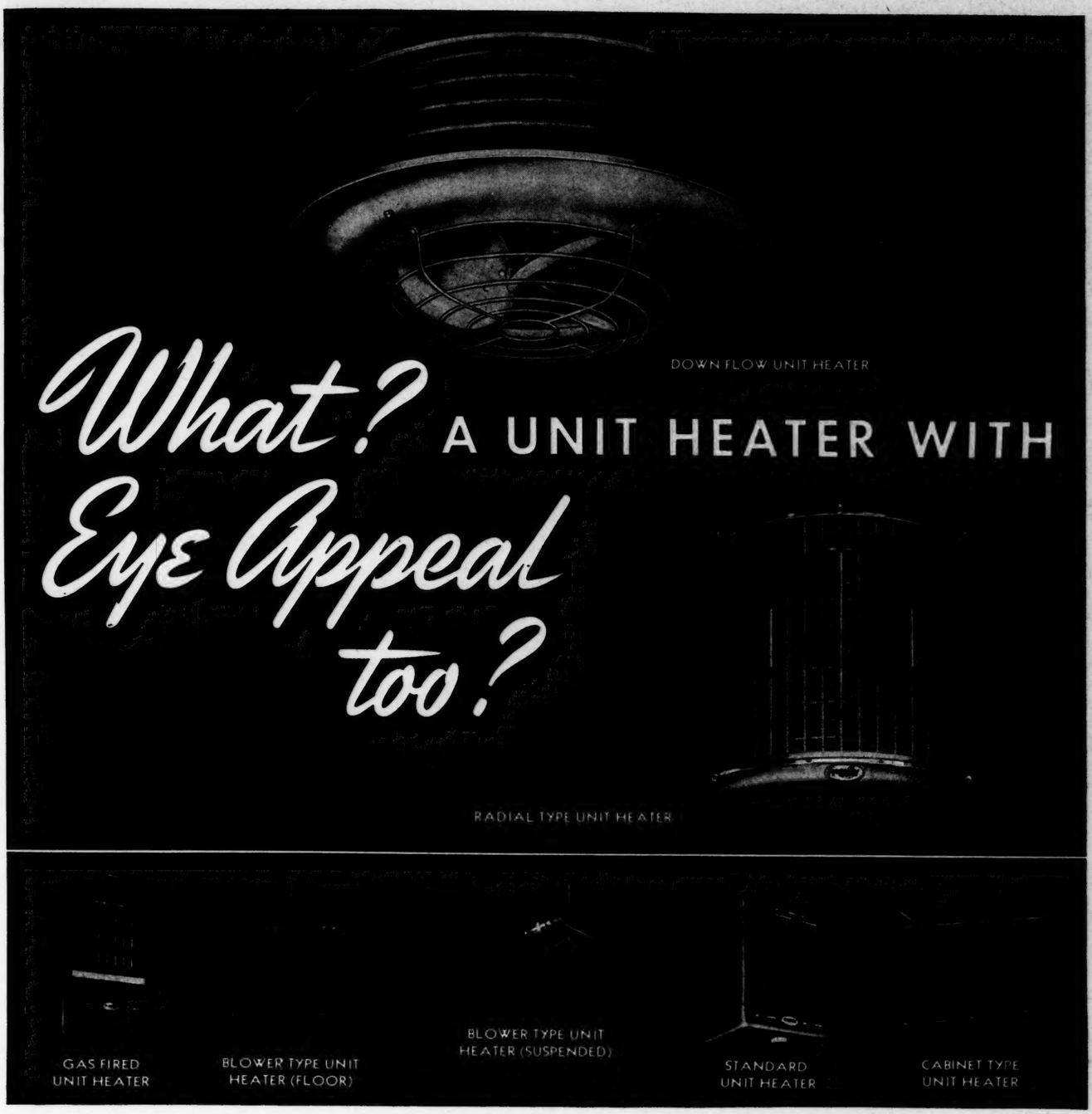
York Ice Machinery Corporation, York, Pa., has appointed *Anker Winther* as assistant general sales manager. Since 1930 Mr. Winther has been connected with York's Cincinnati office as sales engineer and has also served as instructor in their students' training course.

NEW TRADE LITERATURE

Air Distributors. A standard size catalog giving complete information on air distributors made by Plandaire, Inc. Gives information on mounting details, dimensions, method of selection and capacities. PLANDAIRE, INC., P.O. Box 7350, Oakland Station, Pittsburgh, Pa.

Air Outlets. A 4-page, standard size catalog, F-524, describing the model SR Kno-Draft high velocity, spun aluminum ceiling outlet. This device is a combination

(Continued on Page 60)


DOWN FLOW UNIT HEATER
What? A UNIT HEATER WITH
Eye Appeal too?
RADIAL TYPE UNIT HEATER
GAS FIRED UNIT HEATER
BLOWER TYPE UNIT HEATER (FLOOR)
BLOWER TYPE UNIT HEATER (SUSPENDED)
STANDARD UNIT HEATER
CABINET TYPE UNIT HEATER

NO longer do Unit Heaters have to be unsightly. The handsomely styled McQuay Line has real eye appeal, and is complete—a unit heater for every application.

The McQuay Down Flow reclaims wasteful "stratified heat" and forces warmed air down into the working zone over a wide area.

The new Radial Type, an exclusive McQuay development, provides wide uniform heat distribution, lower installed cost and new beauty. One Radial Heater can be used in place of two or more standard units effecting important savings in piping expense as well.

Other McQuay Unit Heaters also have eye appeal combined with the proved McQuay efficiency and long life.

The McQuay catalog describes this complete line and shows you its many definite advantages . . . . Write McQuay, Inc., 1619 Broadway Street, Minneapolis, Minn. Representatives in all principal cities . . .

McQuay INC.

Air Conditioners . . . Air Conditioning Coils . . . Blast Coils . . . Blower Coolers . . . Comfort Coolers . . . Cabinet Radiation . . . Concealed Radiation . . . Evaporative Condensers . . . Indoor Cooling Towers . . . Ice Cube Makers . . . Icy-Flo Accumulators . . . Refrigerating Coils . . . Room Coolers . . . Unit Heaters . . . Unit Coolers . . . Water Cooling Units

and QUAKER STATE finds all 3 STERLING PUMPS are TROUBLE FREE!

Back in the spring of 1933 we put in the first Sterling Deep Well Turbine for the Quaker State Oil Refining Corporation plant near Bradford, Pa. It was a 10" 4 stage unit employing 25 H.P. motor. Later a small unit employing a 3 H.P. motor was installed.

So reliable, so low in maintenance costs and so trouble-free were these two turbine pumps that in November, 1937, Quaker State ordered a third Sterling — a 10" 7 stage pump with 75 H.P. motor. And again Sterling's *free floating* drive shaft and *precision assembly* have given Quaker State savings plus trouble-free operation.

If you want to make savings in handling water, or if you have water handling problems, we can help you. Write us about your problems — today.

Few Sterling pump owners need service—but if you need it, Sterling gives service from coast to coast.

PRECISION BUILT

STERLING PUMP CORPORATION
Hamilton, O. Stockton, Cal.

(Continued from Page 58)

supply and return outlet. Gives information on mounting details and dimensions. PLANDAIRE, INC., P. O. Box 7350, Oakland Station, Pittsburgh, Pa.

Convectors. A standard size, 8-page bulletin, illustrating and describing the Commodore Heater line of Convectofin. Describes the standard Convectofin which is designed for vapor, vacuum systems, the Simplex Convectofin for straight steam with single pipe connections and the Waterflo Convectofin for hot water installations. Capacity tables are given for one and two pipe steam systems. Hot water tables are given in terms of 1000 B.t.u. units. Also contains information on dimensions, cabinet types and construction. COMMODORE HEATERS CORPORATION, 11 West 42nd St., New York City.

Cooling Surface. A standard size, 4-page bulletin on the Nesbitt continuous tube cooling surface. Bulletin shows how the Nesbitt coil solves the problem of draining water from all the tubes to prevent dangerous and costly freezeups. Also includes a number of diagrams including the application of the drain feature to Nesbitt Series W surface. JOHN J. NESBITT, INC., Holmesburg, Philadelphia, Pa.

Draft Gages. A standard size, 4-page bulletin on the Dwyer transparent, stationary draft gage. Lists outstanding features, describes typical applications and gives installation data. F. W. DWYER MANUFACTURING CO., 565 West Washington Blvd., Chicago, Ill.

Dust Collector. A standard size, single page circular No. 42, describing the new American assembled type "Dustube" collector which is recommended where relatively small volumes of air from 17 to 240 c.f.m. are to be handled. Equipment is suited for the collection of dusts created by sand blasting, tumbling, grinding, and similar processes. THE AMERICAN FOUNDRY EQUIPMENT COMPANY, Mishawaka, Indiana.

Fans. A standard size, 14-page catalog No. 40B on the complete line of Airmaster fans. Includes information on attic ventilating systems, commercial exhaust fans, residence ventilators, air circulators, and kitchen ventilating cabinets. AIRMASTER CORPORATION, 4317 Ravenswood Avenue, Chicago, Ill.

Heating Coils. A set of four standard size time saver charts for the quick accurate figuring of temperature rises for various operating conditions with Fedders type K heating coil. Charts are produced in a four-page form on heavy stock, protected by a celluloid-like finish. AIR CONDITIONING DIVISION, FEDDERS MANUFACTURING CO., Buffalo, N. Y.

Heating Systems. A standard size, 12-page catalog, B-99, which describes Taco specialties for warm water heating. Gives information on the "Taco-One" Venturi System, circulators, venturi fittings, two-pipe systems, flow check, expansion tank, relief and reducing valves and controls. Of particular interest is a table which gives in a simple form complete information on how to size the average two pipe, forced circulating warm water job. TACO HEATERS, INC., 342 Madison Ave., New York City.

Humidity. A pocket-size booklet, summarizing in a non-technical manner the dangers of dry air. Points out the effects of dry winter air on furniture, plaster, wall paper, books and plants and gives information on the Harry Alter line of Humidome humidifiers and air cleaners. Booklet should be of help in showing cus-

tomers the advantages of winter humidification. THE HARRY ALTER COMPANY, 1728 So. Michigan Ave., Chicago, Ill.

Motors. A standard size, 4-page bulletin M-50, describing Howell Multi-Speed motors. Gives information on applications for these motors and tells how they save on operating and initial cost. HOWELL ELECTRIC MOTORS COMPANY, Howell, Mich.

Pumps and Traps. Standard size, eighty-page catalog, entitled, "Steam, Air and Water Equipment." Catalog is divided into five sections as follows: (1) Johnson rotary pressure joints, (2) Johnson pressure equalizing boiler feed pumps, (3) Johnson Electrap electrically operated return traps, (4) Johnson boiler water level controls, (5) Johnson gas, air and steam separators, oil absorbers and after-coolers. JOHNSON CORPORATION, Three Rivers, Michigan.

Steam Trap. A standard size two-color, eight-page bullet n 2935 on Cochrane dischargers for high capacity, high pressure drainage. Gives information on the construction, operation, advantages and capacities of these dischargers. Also lists a number of typical installations. COCHRANE CORPORATION, 17th and Allegheny Streets, Philadelphia, Pa.

Thermostats. A standard size, four-page bulletin on Edison sealed thermostats. These thermostats are of the single pole, single throw type and are electrically sealed in an inert, arc quenching gas. Gives information on the advantages and applications of these thermostats. THOMAS A. EDISON, INC., West Orange, N. J.

Thermostats. A 16-page booklet entitled, "A Heated Question Answered," which presents information from a variety of independent sources in answer to the question, "Is there a fuel saving when lower temperatures are maintained at night?" The answer is presented in a concise, yet comprehensive form, which can be easily digested by manufacturers, jobbers, dealers and consumers. MINNEAPOLIS-HONEYWELL REGULATOR COMPANY, Minneapolis, Minn.

V-Belts. Standard size 112-page Vee-Drive catalog V-500, giving complete information on Pyott Vee-Tex drives. Catalog contains all necessary data for selecting the proper drive for any job and includes information on sheave and V-belt specifications, engineering data for design of drives, engineering and mathematical tables, and price information. Also includes photographs of representative drive installations. PYOTT FOUNDRY AND MACHINE COMPANY, 328 North Sangamon Street, Chicago, Ill.

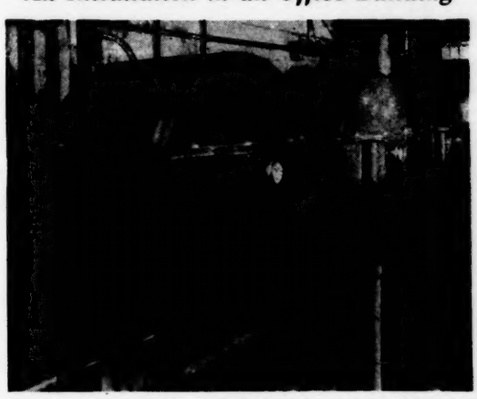
Valves. A standard size, 16-page booklet entitled Reading-Pratt & Cady Bar Stock Valves. Contains complete information, dimensions and application recommendations for bronze, carbon steel, and stainless steel bar stock valves. These valves are machined directly from bar stock metals and are designed to fulfill the need for close control valves capable of withstanding a wide range of temperatures and pressures. READING-PRATT & CADY DIVISION OF THE AMERICAN CHAIN AND CABLE CO., INC., Reading, Pa.

Water Heating. A standard size catalogue, A-99, on Taco Heaters for hot water supply. This catalogue illustrates, describes and gives complete data on the full line of Taco water heaters, both storage and tankless types. TACO HEATERS, INC., 342 Madison Ave., New York City.

GET ACCURATE STEAM COSTS FOR HEATING, PROCESSING WITH THE

ADSCO ROTARY CONDENSATION METER

An Installation in an Office Building



A view of three No. 12,000 ADSCO Rotary Meters in the Chicago Daily News Bldg., Chicago, Ill.

Available from stock in 7 sizes.
250-12,000 lbs. per hour capacity

Totalizes Steam Consumption per Building or Department—Helps You Detect Steam Losses and Improve UTILIZATION

The ADSCO Rotary Condensation Meter is the low cost answer to establishing accurate costs for steam heating per building or department. It is low in first cost, requires little maintenance beyond periodic cleaning, is easy to install, operates under vacuum or gravity conditions and is factory tested to an accuracy within 1%.

Hundreds of these meters are standard equipment on the lines of district heating companies to provide the basis for monthly billing and are used by leading industrial plants, colleges, institutions, federal, state and municipal departments.

Write for quotation giving maximum pounds of hot condensate per hour that the meter would be required to handle or the square feet of equivalent direct radiation in the building or department to be metered.

Write for illustrated Bulletin No. 35-80V.

AMERICAN DISTRICT STEAM COMPANY

NORTH TONAWANDA N.Y.

IN BUSINESS OVER SIXTY YEARS

COMING EVENTS

OCTOBER 7-10. 22nd Annual Convention of the American Gas Association, to be held at The Hotel Traymore, Ritz-Carlton Hotel, Hotel Ambassador and the Atlantic City Auditorium at Atlantic City, N. J. Managing Director, Alexander Forward.

OCTOBER 8-11. The 69th annual meeting of the American Public Health Association, Detroit, Book-Cadillac Hotel, Headquarters.

OCTOBER 9-11. Fall meeting of the Refrigeration Equipment Manufacturers Association convention to be held at French Lick Springs, Indiana. For information write Secretary McClure, REMA Headquarters, 111 West Washington Street, Chicago, Ill.

OCTOBER 14-15. Fall Meeting of the Houston ASHVE at Houston, Texas, to be held under the auspices of the South Texas Chapter and with the North Texas Chapter Members cooperating.

DECEMBER 2-7. 14th National Exposition of Power and Mechanical Engineering at the Grand Central Palace, New York, N. Y. For information write Charles F. Roth, president, International Exposition Company, Grand Central Palace, New York.

DECEMBER 3-5. Winter meeting of the American Society of Refrigerating Engineers to be held at the Hotel Commodore, Lexington Avenue and 42nd Street, New York, N. Y. For information write David L. Fiske, Secretary, ASRE, 37 West 39th Street, New York.

JANUARY 13-16, 1941. Third Refrigeration and Air Conditioning Exposition to be held at Stevens Hotel, Chicago.

JANUARY 27-29, 1941. 47th Annual Meeting of the American Society of Heating and Ventilating Engineers to be held at Hotel Muehlbach, Kansas City, Missouri, with Kansas City as Host Chapter.

MARCH 17-22, 1941. National Oil Burner Progress Exhibition of oil heating and air conditioning equipment. Commercial Museum, Philadelphia. Further information available through C. F. Curtin, Oil Burner Institute, 30 Rockefeller Plaza, New York.

JUNE 16-20, 1941. The Pacific Heating and Air Conditioning Exposition, Exposition Auditorium, San Francisco, Cal. Managed by the International Exposition Company, Grand Central Palace, New York.

JUNE 16-20, 1941. Meeting of the Heating, Piping and Air Conditioning National Association, San Francisco, Cal.

JUNE 16-20, 1941. Summer meeting of the American Society of Heating and Ventilating Engineers, to be held in the Civic Center Auditorium, San Francisco, Cal.

JUNE 23-27, 1941. Annual meeting and exhibit of the American Society for Testing Materials. Palmer House, Chicago, Ill. For information, write Society at 260 So. Broad St., Philadelphia, Pa.

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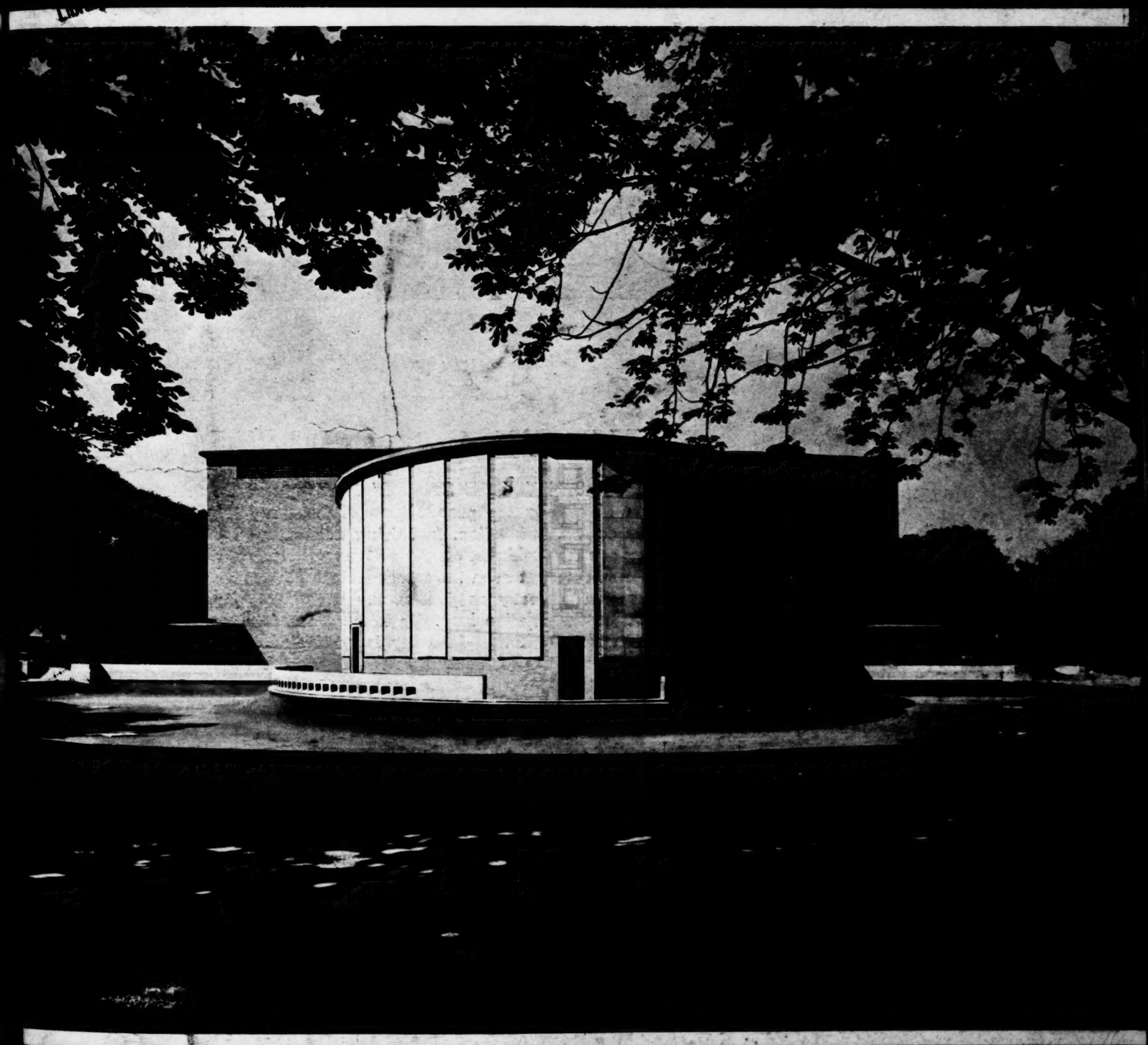
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